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Inflatable Vertical Float System

Phase I Report

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MARTIN MARIETTA

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Inflatable Vertical Float System.

Phase I Report,

10

~~Prepared by:~~ J. E. Pauken and R. E. Rieckmann

~~Approved by: [illegible]~~

15 N00019-67-C-0289

14 ER-14680

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12 82p.

MARTIN MARIETTA CORPORATION
BALTIMORE, MD. 21203

(403 251)

CONTENTS

1.0 SUMMARY

This report summarizes the work done under Phase I of a proposed three phase program to engineer and demonstrate ~~an~~ "Inflatable Vertical Float System" (Navy Contract N00019-67-C-0289). The results of this study indicate that the design concept, as developed by the Martin Marietta Corporation in conjunction with Uniroyal, Inc., represents a feasible lightweight Inflatable Vertical Float System. Phase II of the proposed program would consist of complete engineering of the design for the HRV airplane, in conjunction with Thurston Aircraft Corporation, and the construction and demonstration testing of a single Inflatable Vertical Float with representative systems. The Phase III effort would consist of outfitting the HRV airplane with the Inflatable Vertical System and performing full scale demonstration tests in open ocean environments.

This proposed program will put to practical demonstration the concept of seaplane open ocean ASW operation. ~~(References 18 and 19).~~ The HRV is an approximate 1/3 ~~model~~ scale representation of the P5 seaplane. It is equipped with a hydrofoil and has satisfactorily demonstrated take off and landing in seas three times the wave height permissible with the basic hull. ~~(Reference 15)~~. Then with the incorporation of the Inflatable Vertical Float System, the HRV will have open ocean seakeeping capability and thus be a complete open ocean seaplane. The overall float system is shown. ~~in Figure 1.1.~~

Initially, design goals were established and all were achieved within the scope of the study. These were, a system weight of 400 lbs or less, a 30 second retraction or extension cycle, capability to move through the water at 6 knots, identical float construction and size, and inflation system capability for two deployment-retraction cycles.

The Inflatable Vertical Float System is made up of a central telescoping column which supports an accordian type flotation bag. Local intermediate ribs are provided to maintain the cross sectional shape and take any torsional loads. This can be seen in Figure 1 and the details in Drawing No. 818172. The HRV airplane with the Vertical Float System is shown in the ~~overall configuration~~ drawing (No. 818171).

Extension of the Inflatable Vertical Float System is accomplished by a single 2000 psig nitrogen storage bottle and associated pneumatic system. Control is provided at the pilot station through two flow control valves which regulate wing and hull float inflation. The operating pressure for the Inflatable Float System is 5 psig.

The following sections discuss in detail the studies ~~which were~~ conducted and their results as related to the design of the Inflatable Vertical Float System.

accordion

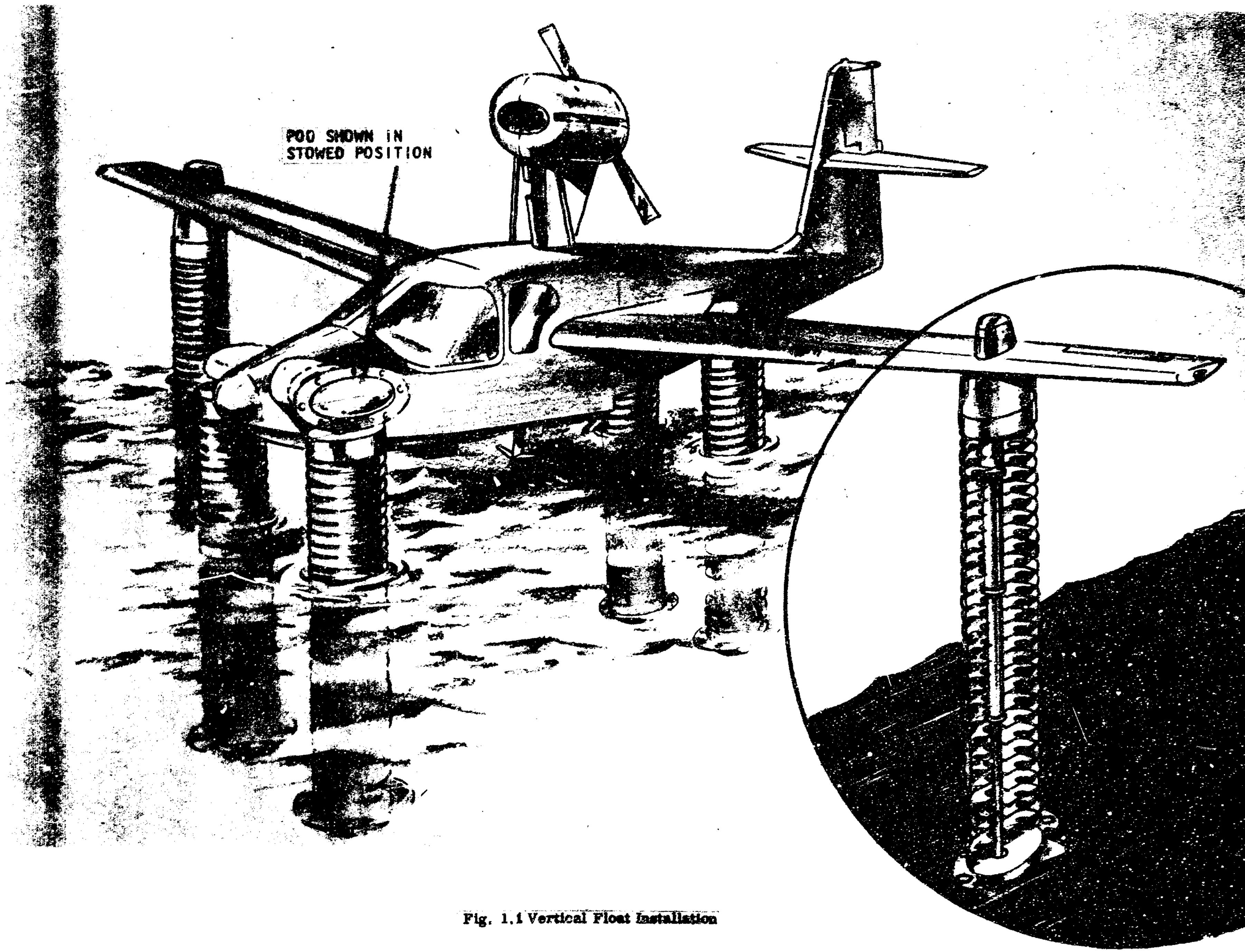
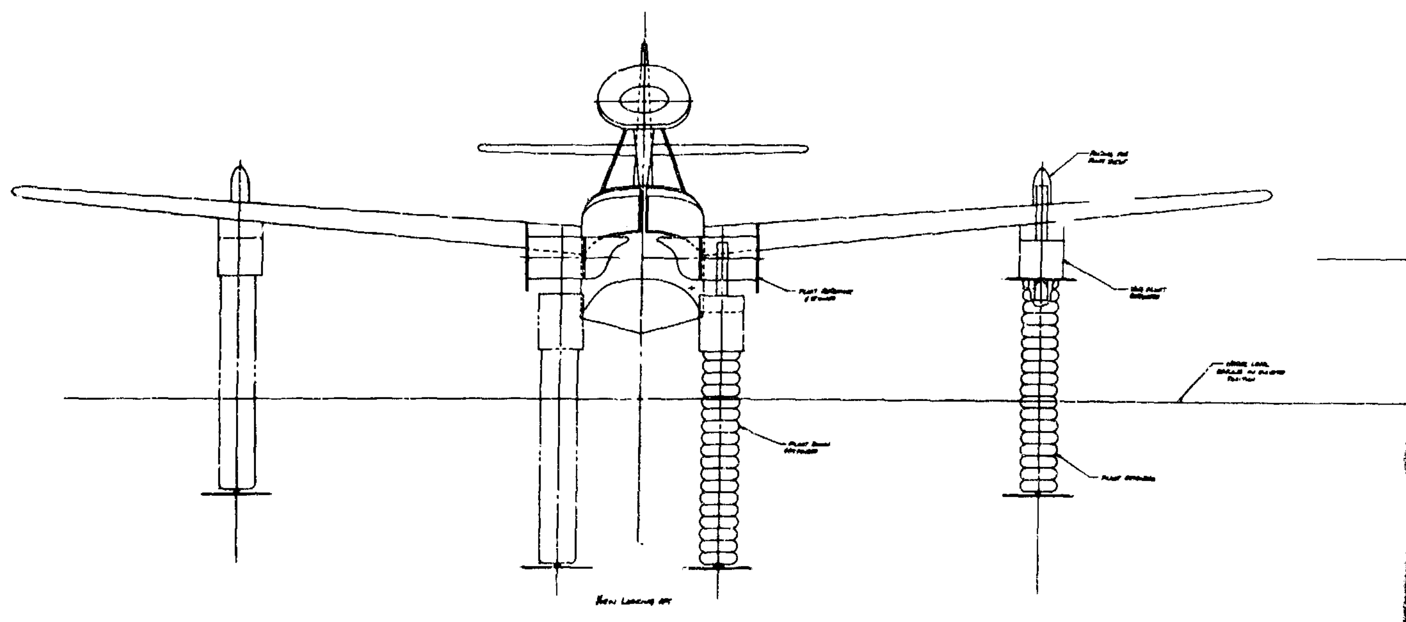
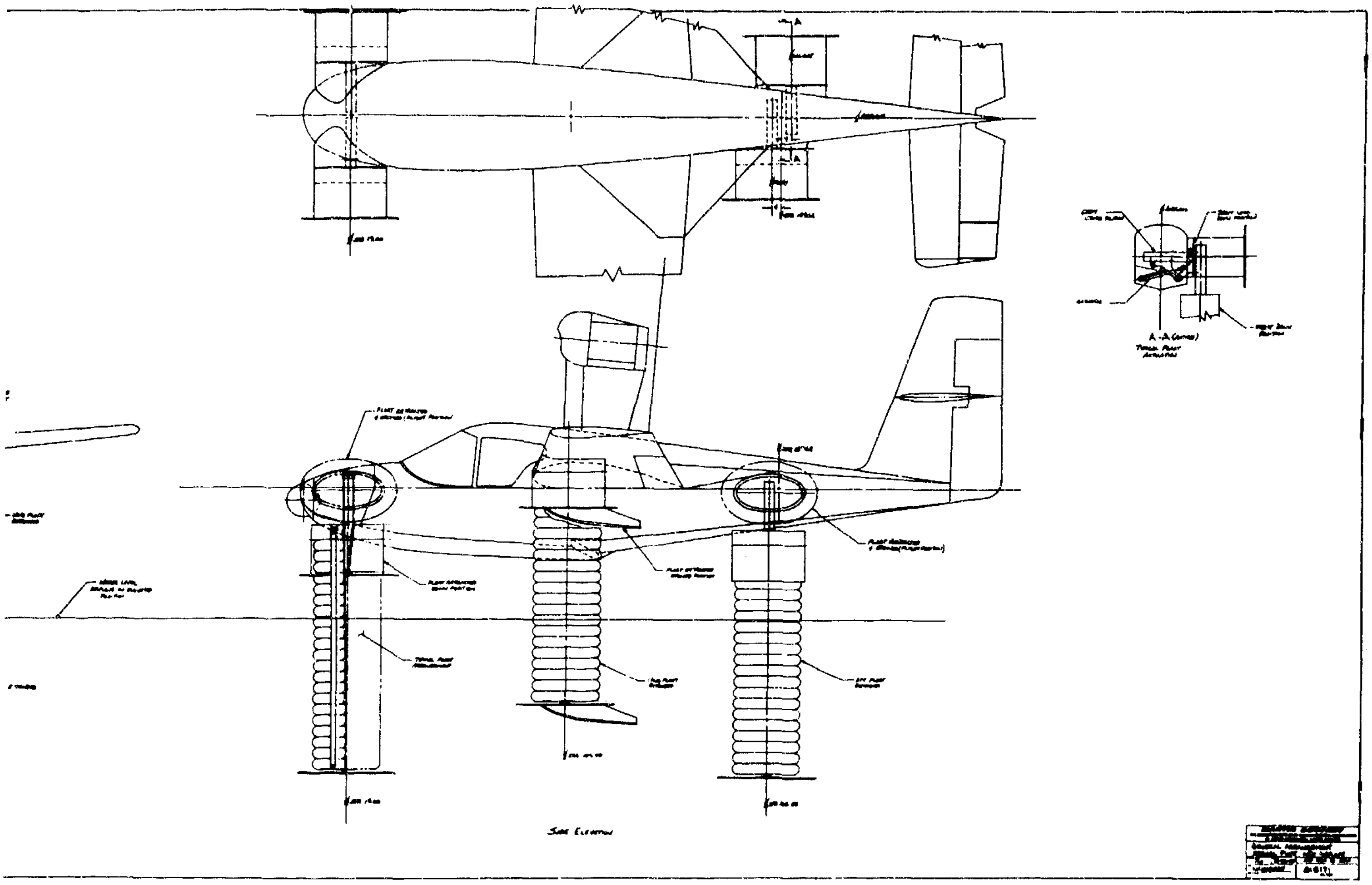
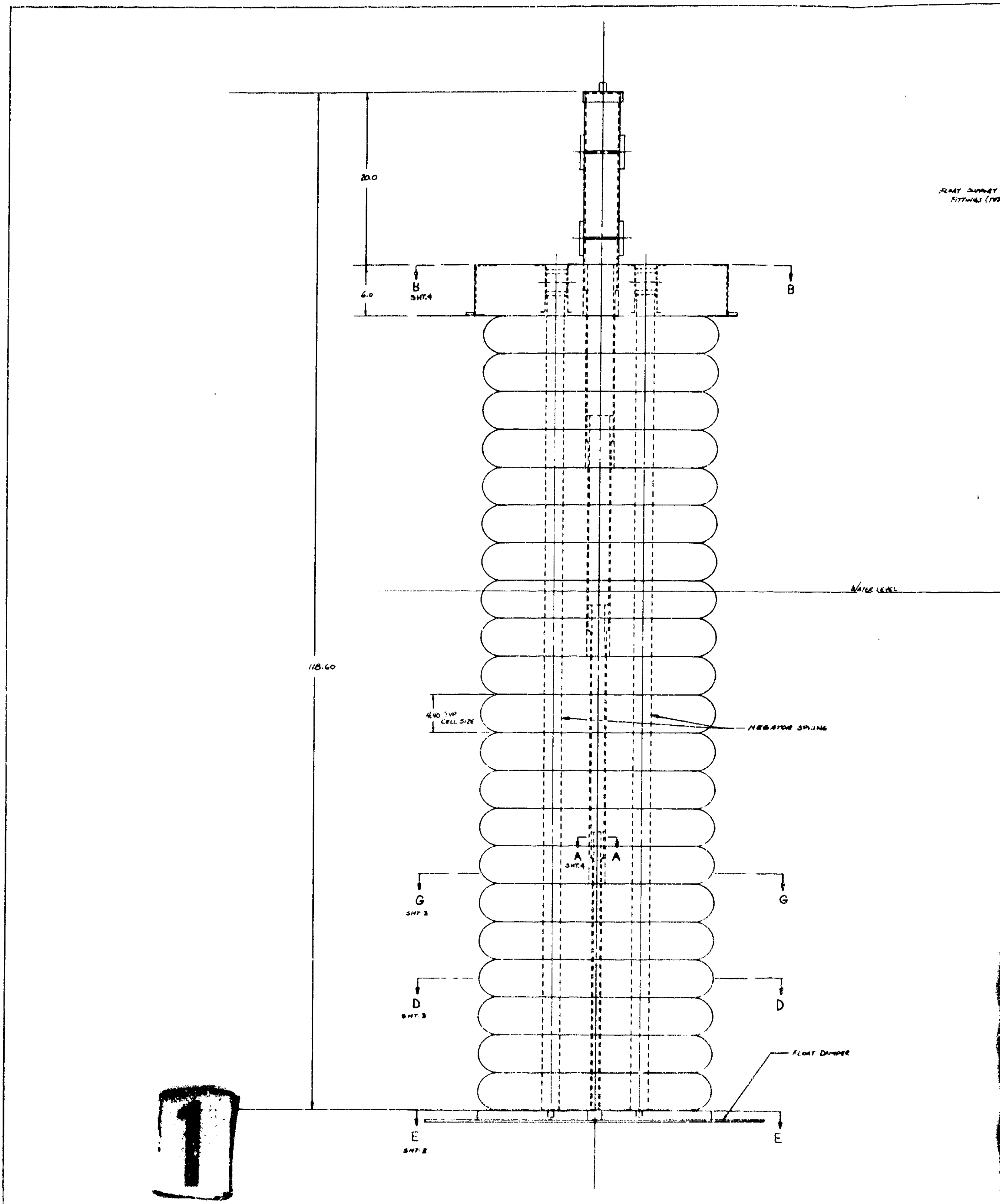


Fig. 1.1 Vertical Float Installation






20.0
6.0
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NEGATOR SPRING
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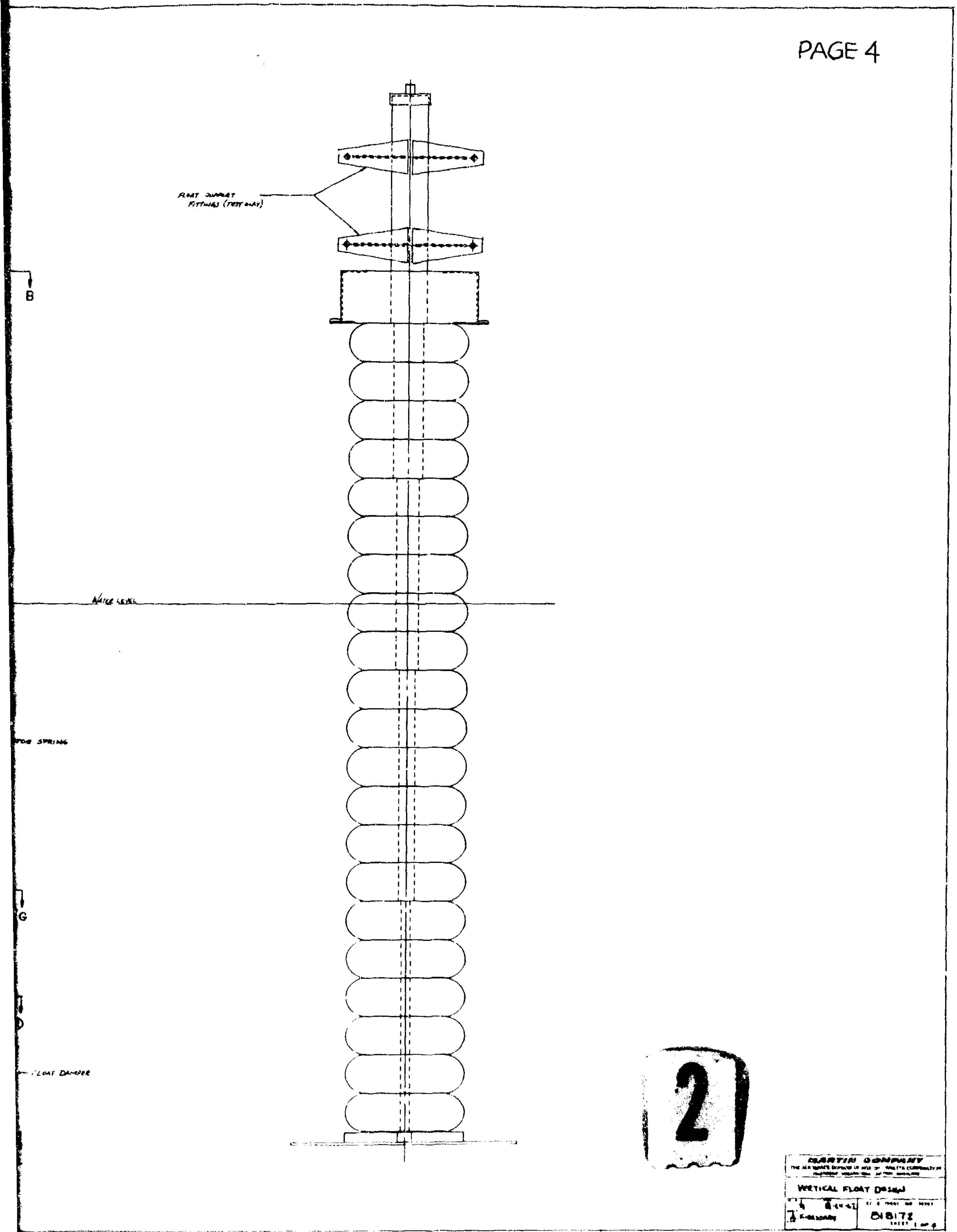
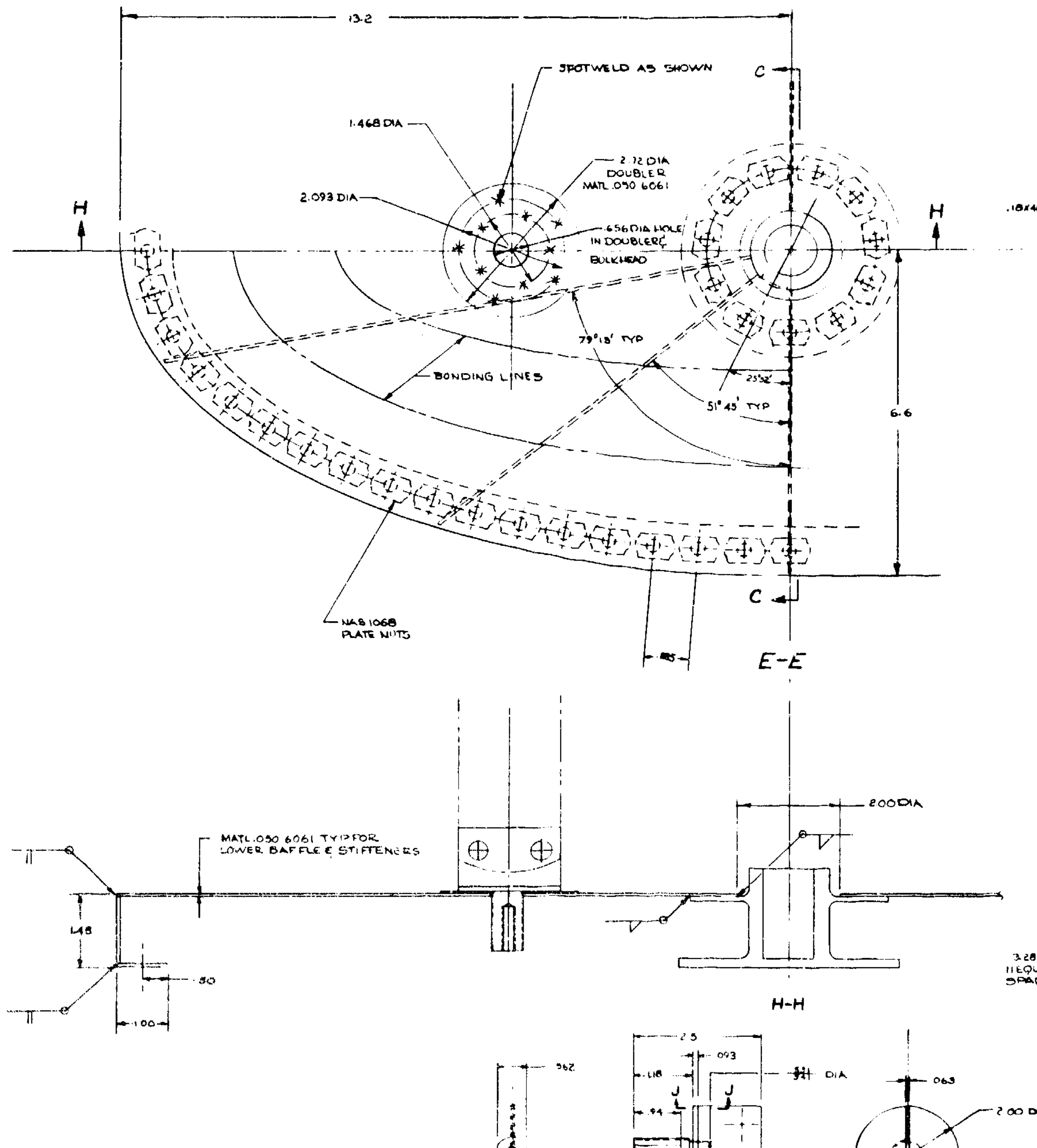
SHT.2
E

1


FLOAT SUPPORT
FITTINGS (TEST ONLY)
B
WATER LEVEL
G
2
MARTIN COMPANY
VERTICAL FLOAT DESIGN
SHEET 1 OF 4



13.2
SPOTWELD AS SHOWN
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1.468 DIA
2.12 DIA
DOUBLER
MATL .050 6061
2.093 DIA
H
H
.656 DIA HOLE
IN DOUBLER
BULKHEAD
79°18' TYP
BONDING LINES
25°52'
51°45' TYP
6.6
NAS 1068
PLATE NUTS
C
E-E
2.00 DIA
MATL .050 6061 TYP FOR
LOWER BAFFLE & STIFFENERS
1.48
.50
1.00
H-H
2.5
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DIA
.063
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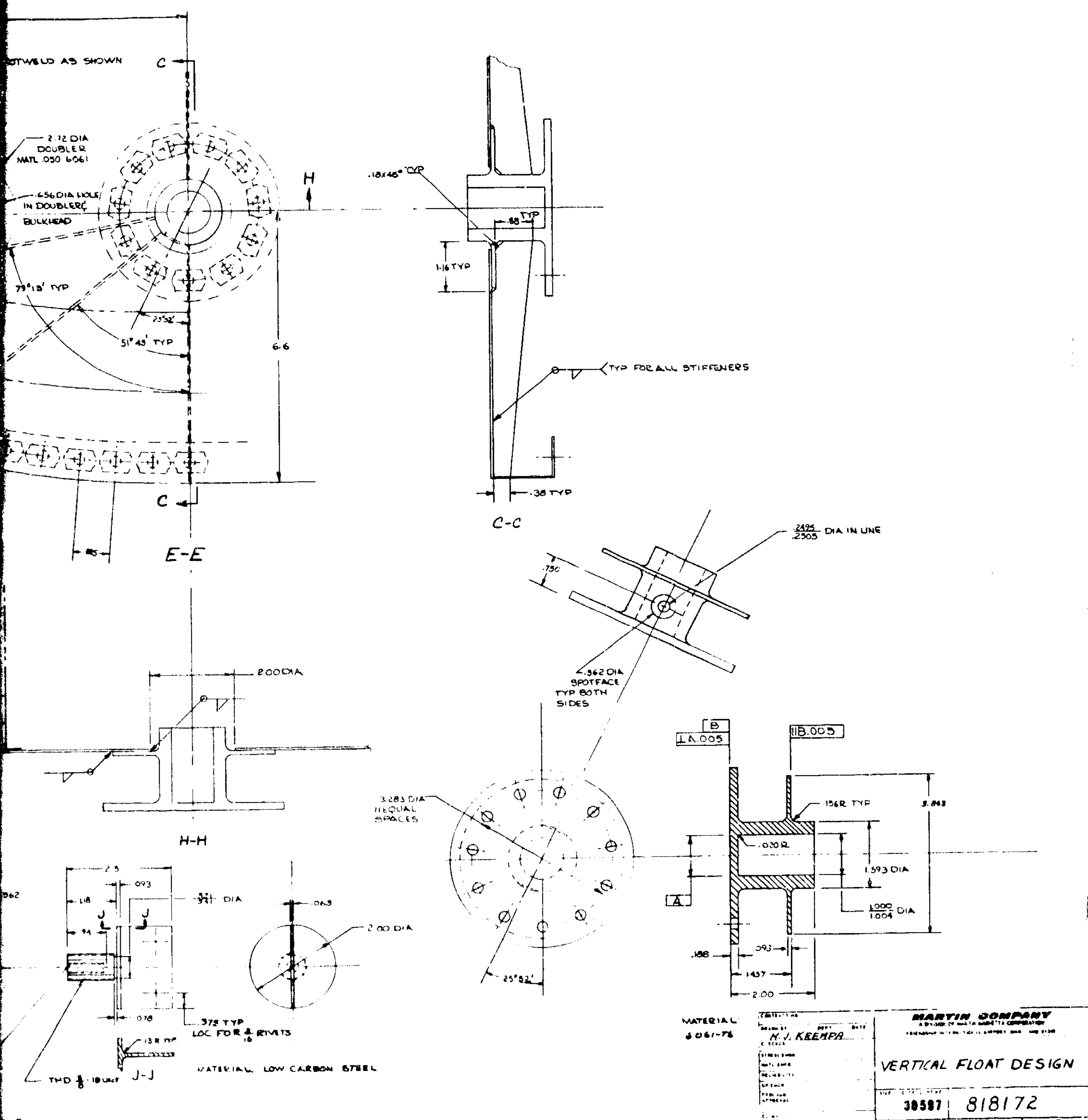
LOC FOR 3/16 RIVETS
.13 R TYP
J-J
THD 5/8-18 UNF
MATERIAL: LOW CARBON STEEL
THD 1/4-28 UNF

1



SPOTWELD AS SHOWN
2.12 DIA DOUBLER MATL .050 6061
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79°13' TYP
25°52'
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E-E
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.88 TYP
1.16 TYP
TYP FOR ALL STIFFENERS
.38 TYP
C-C
.2495 / .2505 DIA IN LINE
.750
.562 DIA SPOTFACE TYP BOTH SIDES
2.00 DIA
H-H
3.283 DIA 11 EQUAL SPACES
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A .005
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B .005
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1.593 DIA
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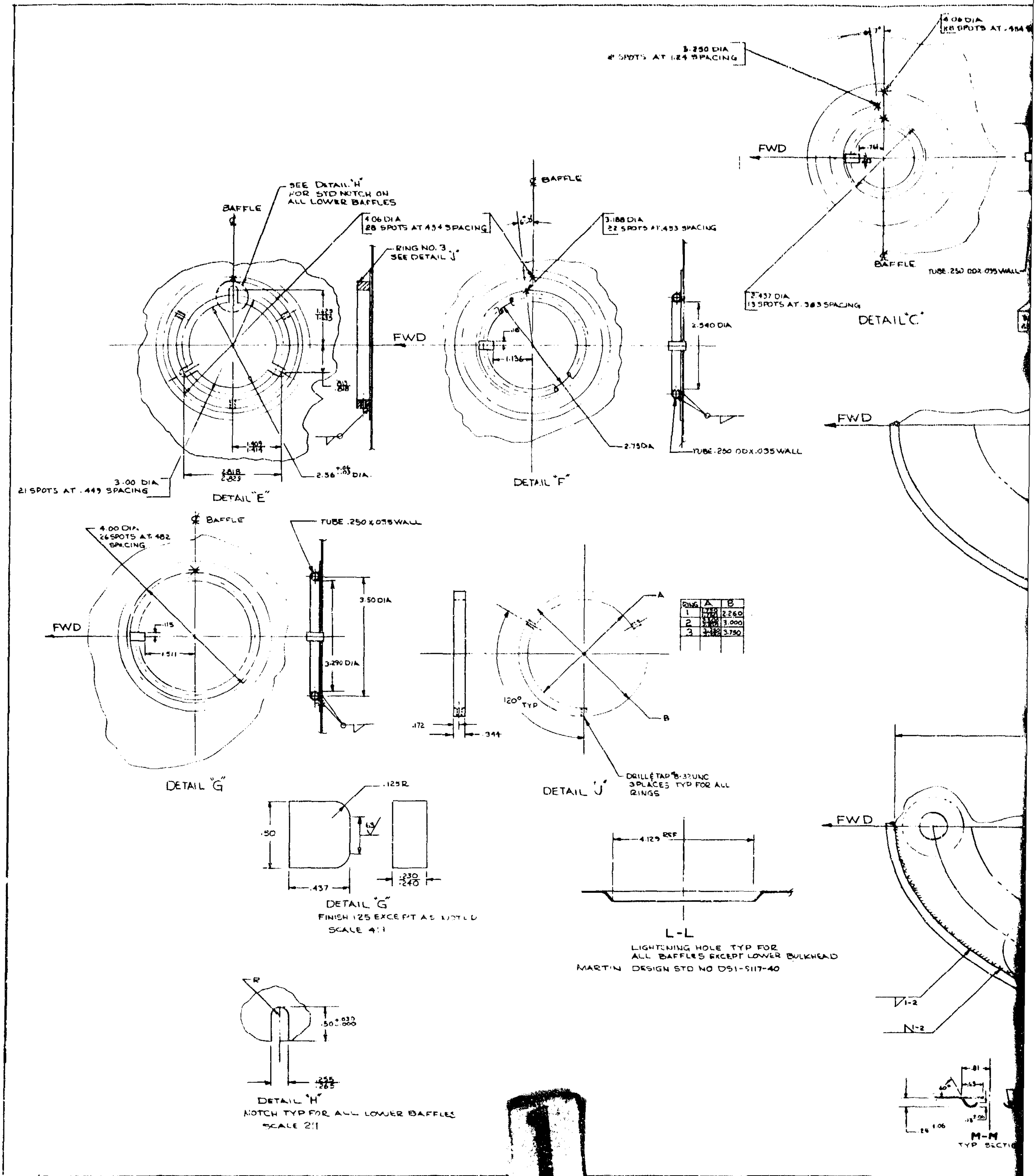
.13 R TYP
THD 1/8-18 UNF
J-J
MATERIAL LOW CARBON STEEL
MATERIAL 6061-T6
M. J. KREMPA
MARTIN COMPANY
VERTICAL FLOAT DESIGN
38597
818172



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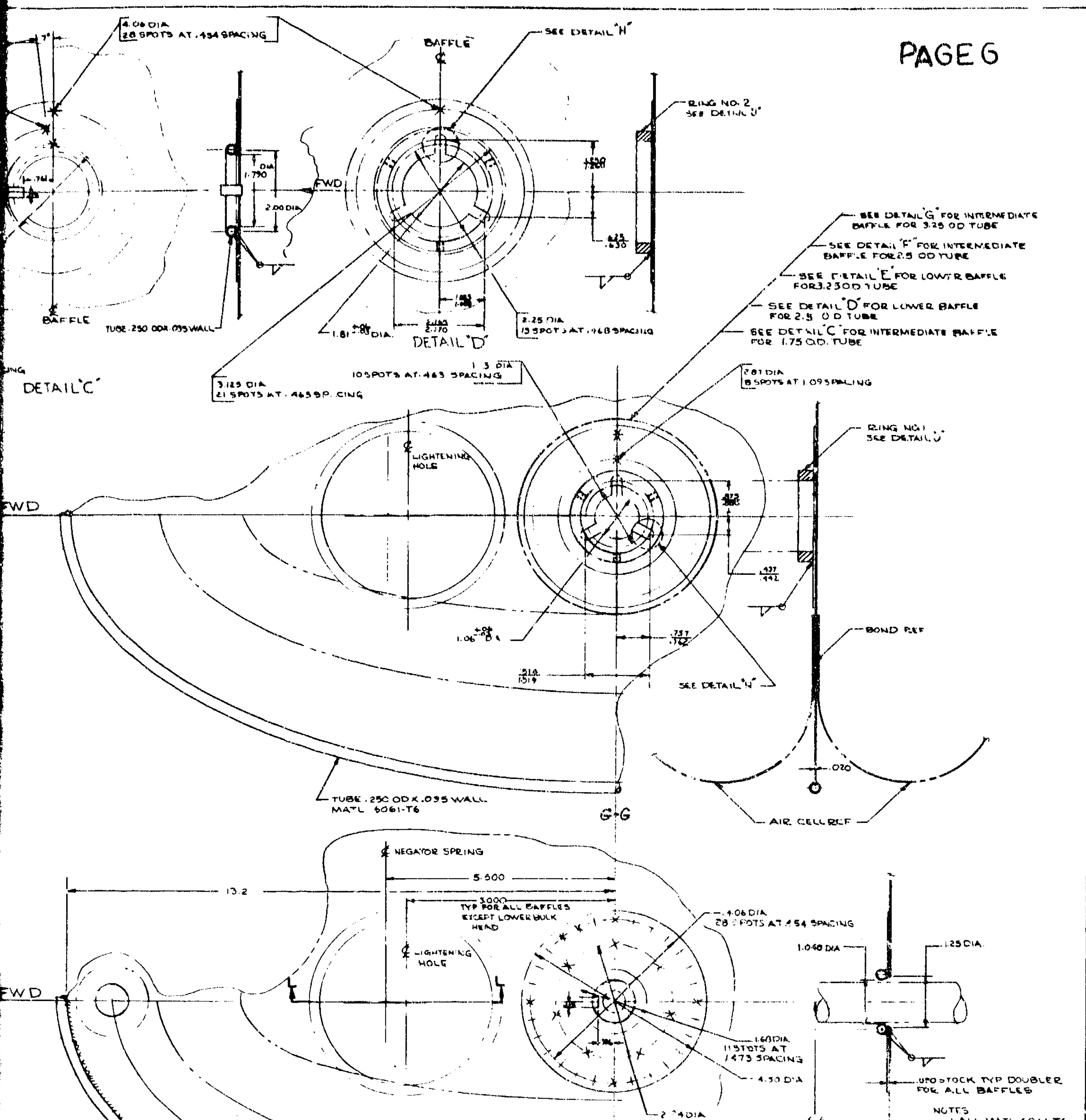
2	[illegible]	3.000
3	[illegible]	3.750



PAGE 6
DETAIL "C"
DETAIL "D"
BAFFLE
FWD
TUBE .250 OD X .035 WALL
SEE DETAIL "H"
RING NO. 2 SEE DETAIL "J"
2.25 DIA 15 SPOTS AT .468 SPACING
3.125 DIA 21 SPOTS AT .465 SPACING
4.06 DIA 28 SPOTS AT .454 SPACING
SEE DETAIL "G" FOR INTERMEDIATE BAFFLE FOR 3.25 OD TUBE
SEE DETAIL "F" FOR INTERMEDIATE BAFFLE FOR 2.5 OD TUBE
SEE DETAIL "E" FOR LOWER BAFFLE FOR 3.25 OD TUBE
SEE DETAIL "D" FOR LOWER BAFFLE FOR 2.5 OD TUBE
SEE DETAIL "C" FOR INTERMEDIATE BAFFLE FOR 1.75 OD TUBE
10 SPOTS AT .463 SPACING
2.81 DIA 8 SPOTS AT 1.095 SPACING
RING NO. 1 SEE DETAIL "J"
LIGHTENING HOLE
BOND REF
SEE DETAIL "N"
AIR CELL REF
TUBE .250 OD X .035 WALL MATL 6061-T6
G-G
NEGATOR SPRING
5.500
13.2
5.000 TYP FOR ALL BAFFLES EXCEPT LOWER BULK HEAD
4.06 DIA 28 SPOTS AT .454 SPACING
1.040 DIA
1.25 DIA
1.60 DIA 11 SPOTS AT .473 SPACING
4.30 DIA
2.74 DIA 7 SPOTS AT 1.25 SPACING
6.6
.020 STOCK TYP DOUBLER FOR ALL BAFFLES
NOTES
1. ALL MATL 6061-T6
2. THE PERIPHERAL TUBES OF THE BAFFLES SHALL HAVE A GAS VENT HOLE & SHALL BE PLUGED AFTER WELDING
3. ALL RINGS & KEYS SHALL BE POSITIONED AS SHOWN
TUBE .250 OD X .035 WALL MATL 6061-T6
SEE DETAIL "G"
.020 STOCK
TUBING .25 DIA X .035 WALL
M-M TYP SECTION FOR ALL BAFFLES
BAFFLE FOR 1 DIA TUBE
D-D
MARTIN COMPANY
VERTICAL FLOAT DESIGN
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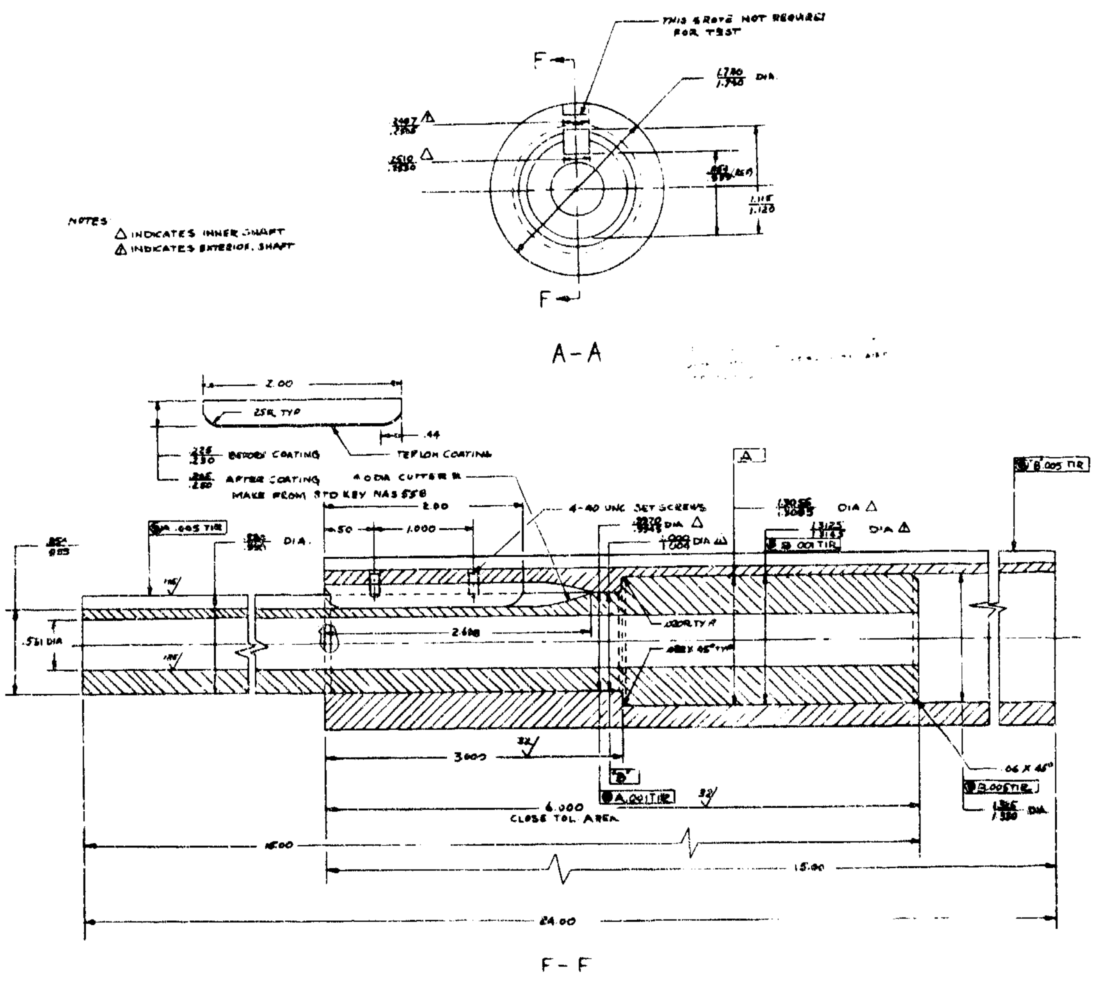
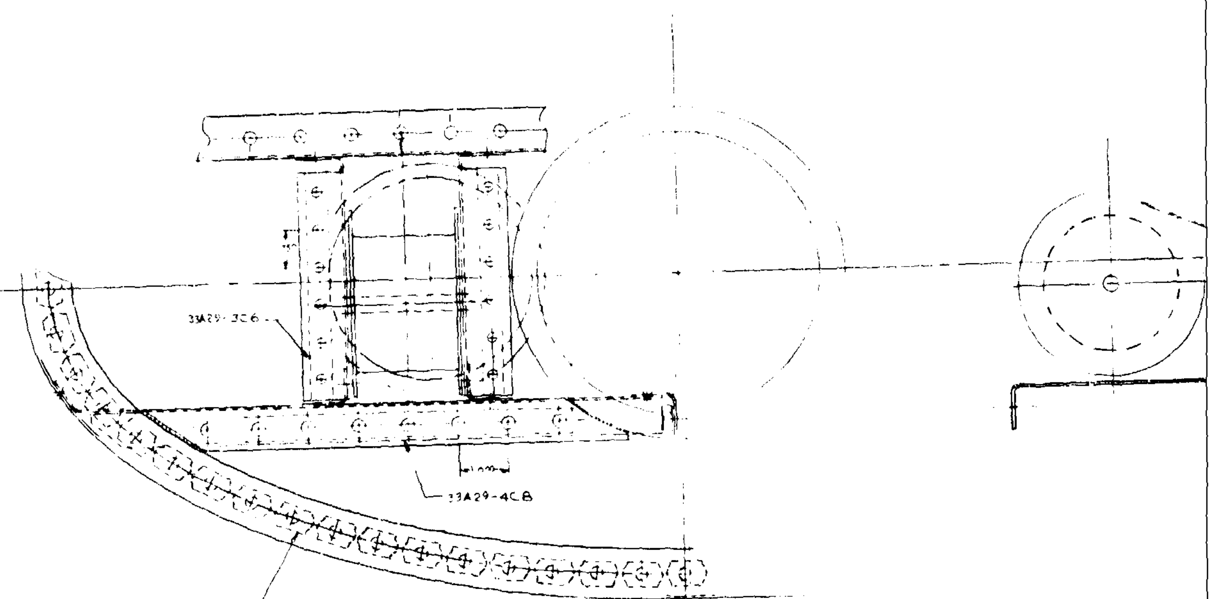

THIS GROVE NOT REQUIRED FOR TEST
NOTES:
△ INDICATES INNER SHAFT
△ INDICATES EXTERIOR SHAFT
A-A
TEFLON COATING
4-40 UNC SET SCREWS
CLOSE TOL. AREA
F-F

1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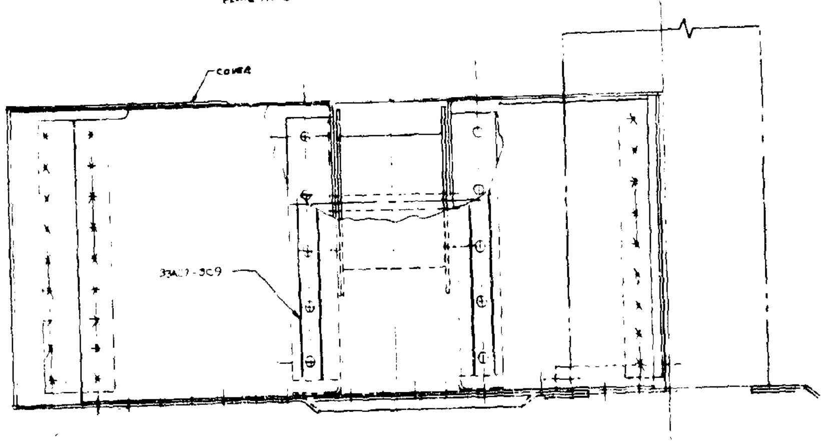
33A29-4C8
NAS 1068
PLATE NUTS

B-B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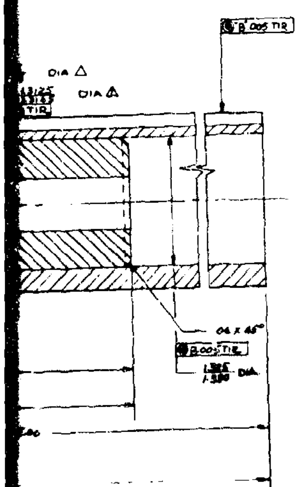
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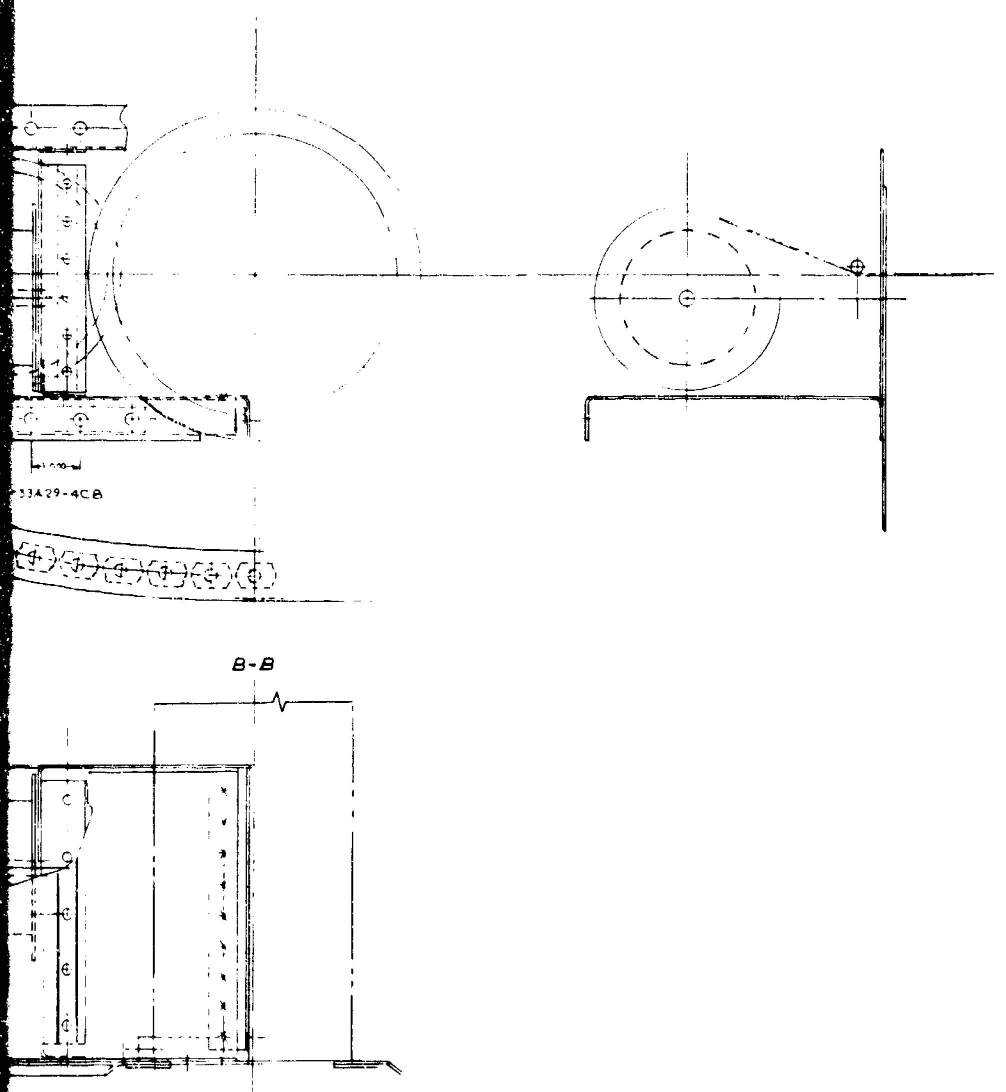
2



3

H J KREMPA

VERTICAL FLOAT DESIGN

818172

2.0 INTRODUCTION

The concept of a Vertical Float System to provide seakeeping capability to a seaplane has had extensive evaluation (References 1, 3 through 10) and testing. All the results show that significant wave isolation is possible for the aircraft thus providing a usable ASW mission platform. The Martin Marietta Corporation, with the assistance of Uniroyal, Inc. and Thurston Aircraft Corporation, therefore, proceeded with the Phase I effort to develop and evaluate the Martin Marietta Vertical Float concept. The Martin Marietta concept offers three principle advantages over previous concepts. These are: the need for flooding the float system with sea water during the extension and retraction cycles is eliminated, thereby eliminating large heavy water pumping equipment and the need for an air cooling or heat exchanger system; the floats can be extended or retracted very quickly; and lastly, it provides simplicity which inherently provides reliability. These advantages are accomplished by separating the structural and flotation functions.

As is the case for many developed systems, the Inflatable Vertical Float concept which was finally employed was the result of many investigations. During the early phases of engineering work, the fluid column approach appeared to offer advantages and development and test work was completed. Appendix A summarizes the fluid column test results. Considering that the fluid column approach required large retraction equipment, a new direction was taken and the present concept evolved. The recommended approach basically separated the flotation and support functions whereas the fluid column utilized a combined flotation-support function.

The overall float system is illustrated in Figure 1.1. This sketch shows the system installed on the HRV airplane for ocean demonstrations. The system and float design was developed in such a manner as to provide minimum modification to the airplane. From Figure 1.1, it can be noted that dual floats are used for the forward and aft hull locations. This was predicated by the ground rule that the amphibian characteristics be maintained for the demonstration airplane.

After the aircraft has landed in the water and becomes stationary, the hull floats are rotated into the extension position by hydraulic power. After rotation, the floats are expanded utilizing air bottles in the aircraft. The air flow will be controlled at the pilot and/or copilot station through the use of manual throttle valves. This will allow adjustment during the lifting cycle and final trim position if necessary. The fully extended position is controlled by built-in stops in the telescoping structural support member. With the aircraft in the fully extended float position, ASW work may progress with but a minimal water surface motion being transmitted to the aircraft platform. The basic float sections, by virtue of their construction, will have inherent damping qualities which are supplemented by additional damper plates at the bottom of the float. For the retraction cycle, the bags are vented through the control valves while the buoyant forces compress the

floats until the aircraft has settled in the water. Two negator springs are utilized in each float to provide a positive force for final retraction when the buoyant forces approach a minimum value.

In order to provide the airplane with realistic operational capability, the following design goals were established and incorporated into the Vertical Float System:

(1) The total Vertical Float System shall not weight more than 400 lbs.

(2) The time required to deploy and retract shall not exceed 30 seconds each.

(3) The float system shall be capable of moving through the water at 6 knots.

(4) All floats will be identical in length and cross section.

(5) Air bottles stowed on board the aircraft may be used as substitute power systems.

(6) The stowed air bottles shall have the capability to provide any power load and two deployment and retraction cycles.

The complete program from the engineering to the actual demonstration is a three phase effort. The first being the study and design evaluation of the float concept, the second phase consists of manufacturing a single float and testing the system concept, and the third phase the engineering, modification, installation and testing of the float system and HRV airplane.

The following sections discuss the results of the Phase I evaluation of the Martin Marietta Vertical Float concept.

3.0 DESIGN STUDIES

3.1 Design Criteria

In order that the Vertical Float concept could be demonstrated on the HRV seaplane, the design criteria used reflected the practical aspects of usable hardware. The HRV aircraft is an approximate 1/3 scale model of the P5A seaplane. Requirements of the full scale seaplane indicated the ability to perform ASW search in sea states up to sea state 4. The recommendations of Reference 1 state that for the P5A aircraft, a clearance of 6 feet would be sufficient. This was partially proven through the open ocean tests of vertical float supported PBM-5 airplane. Then for the third scale demonstration, a minimum keel clearance of 2 feet was used. This would allow a random maximum wave height of 4 feet for the HRV airplane.

It becomes quite conceivable that an aircraft on an ASW mission must have the capability to move to various locations. To provide this capability, the float concept was designed for a 6-knot speed through the water. The aircraft will use its own power for maneuvering. Water speed will be controlled by the pilot's ability to sense speed or trim attitude or through the use of appropriate instrumentation. To provide adequate strength in the primary structure, a factor of safety of 2.0 was used. This would allow a speed margin of 41% or a 2.5-knot tolerance.

The ability of any body to move through the water is dictated by its drag characteristics which are a function of the cross sectional shape and speed. In order to determine the best cross section to use for the floats, a representative number of cross section shapes was evaluated for drag performance. The shapes considered are shown in Table 3.1 along with their critical dimensions. These dimensions reflect the cross sectional area requirements of the HRV airplane, which are approximately 2 sq. ft.

TABLE 3.1

STRUT SECTIONS AND DIMENSIONS USED FOR DRAG STUDY

Shape	Chord	Thickness	Depth	t/c	Reynolds No.
Circle	1.6 ft.	1.6 ft.	4.9 ft.	1.0	1.23×10^6
Streamline	2.64 ft.	1.06 ft.	4.9 ft.	.4	2.03×10^6
Ellipse (2.5:1)	2.54 ft.	1.02 ft.	4.9 ft.	.4	1.95×10^6
Ellipse (2:1)	2.27 ft.	1.13 ft.	4.9 ft.	.5	1.75×10^6

Using the data of Reference 2, the total drag of the various sections was derived. This drag was made up of the following components: friction drag, wave drag, and interference drag. The latter component is due to the use of dual floats at the forward and aft hull stations. Skin friction coefficients were determined with Figure 3.1.1 for the respective Reynolds number. The results of the drag study showed that both the 40% thickness streamline and ellipse had the best drag characteristics. Figure 3.1.2 shows the relative drag characteristics as a function of thickness/chord ratio. As a result of the drag study, the 50% elliptical shape was selected for the float cross section. A study was then completed to determine the strut drag as a function of speed. This is shown in Figure 3.1.3.

The aircraft's ability to move through the water, while the floats are extended, will be a function of the float drag forces and the available thrust. Figure 3.1.3 shows the anticipated drag force per float as a function of speed. Considering the HRV configuration with extended floats, it becomes apparent that a significant nose down pitching moment will exist due to the engine thrust-drag load couple distance. This pitching moment and resulting trim angle may be alleviated by using the stabilizer. An estimate of the stabilizer's effectiveness in the engine slip stream was made by Mr. D. Thurston for the HRV aircraft. With this information, the curve of available stabilizer moment was derived. This is shown in Figure 3.1.4. Having this information, a study was made to determine the trim attitude of the HRV as a function of speed. This is shown in Figure 3.1.5 for no stabilizer moment relief and with stabilizer moment relief. Also shown is the bow impact angle. The 6-knot condition is attainable and, in addition, the pilot will be able to sense a bow impact before the structural capability is exceeded.

In order to meet the requirements of a 30-second retraction or extension cycle and to accomplish this with an air pressure system, it was necessary to consider a pressure source which had high internal pressure. The float construction was configured in such a way as to maximize its damping characteristics. In addition, this damping also minimizes the aircraft response to a seaway.

During the evaluation of the float packages, every effort was made to keep the flight characteristics of the retracted float package as clean as possible with a minimum of airplane configuration change.

3.2 Configuration

In principle, it is desirable to have the vertical floats as deep in the water as possible to minimize the response of the aircraft to water surface perturbations. However, hydrostatically, as the length of the float increases, the water plane moment decreases which causes hydrostatic instability. Keeping this factor in mind, a preliminary float sizing study was made selecting a range of cg height above the float center of buoyancy and float span.

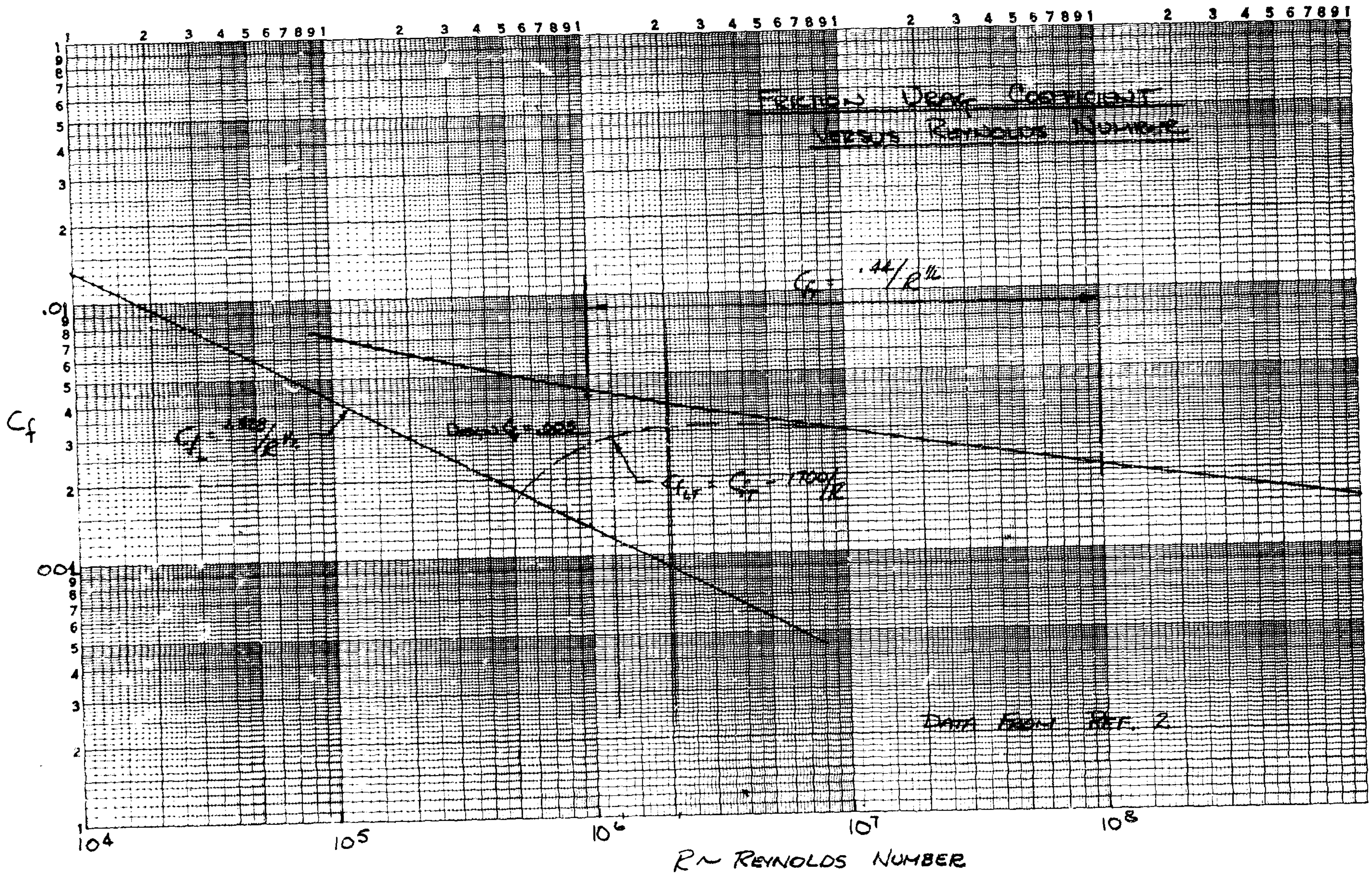


FIGURE 3.1.1 – FRICTION DRAG DATA

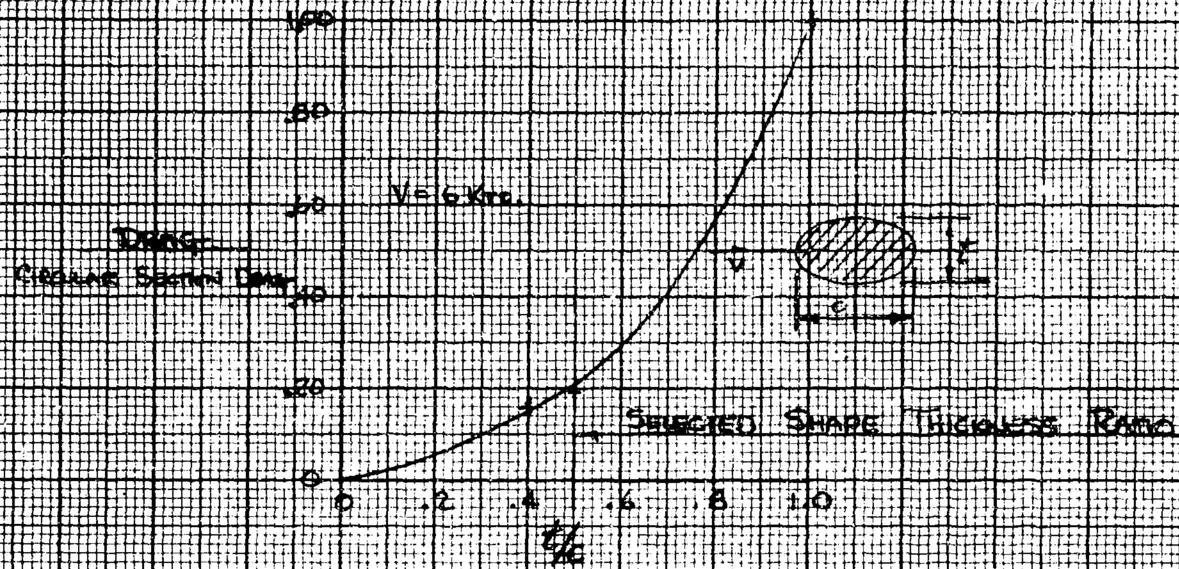


FIGURE 3.1.2- DRAG VARIATION OF VARIOUS STRUT CROSS SECTIONS AS COMPARED TO CIRCULAR CROSS SECTION

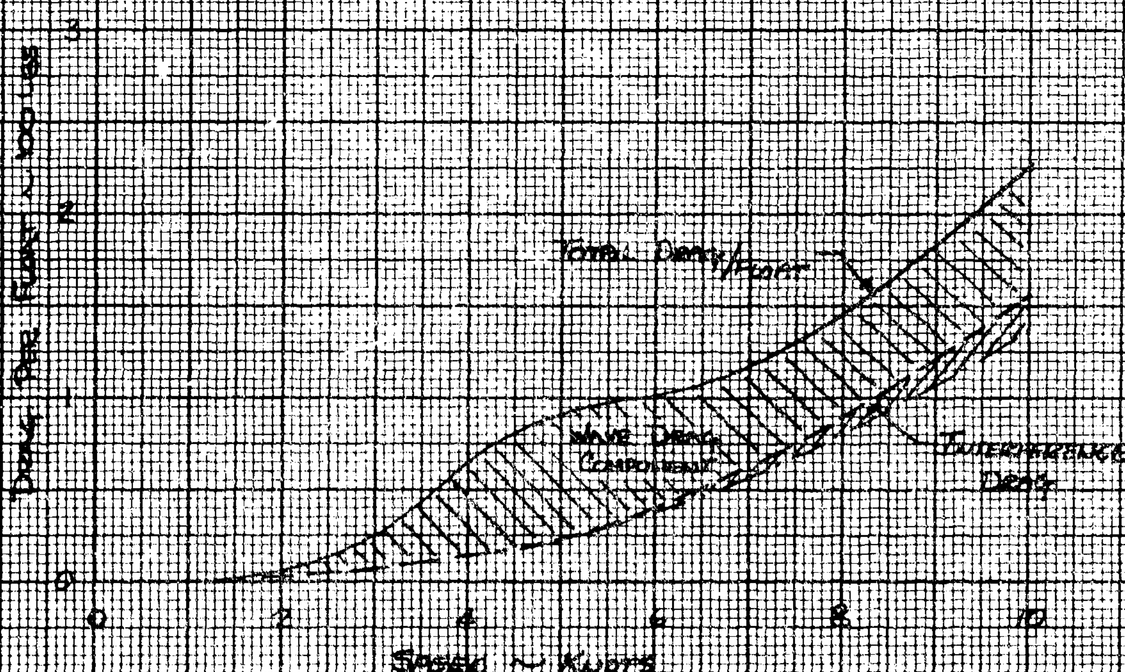
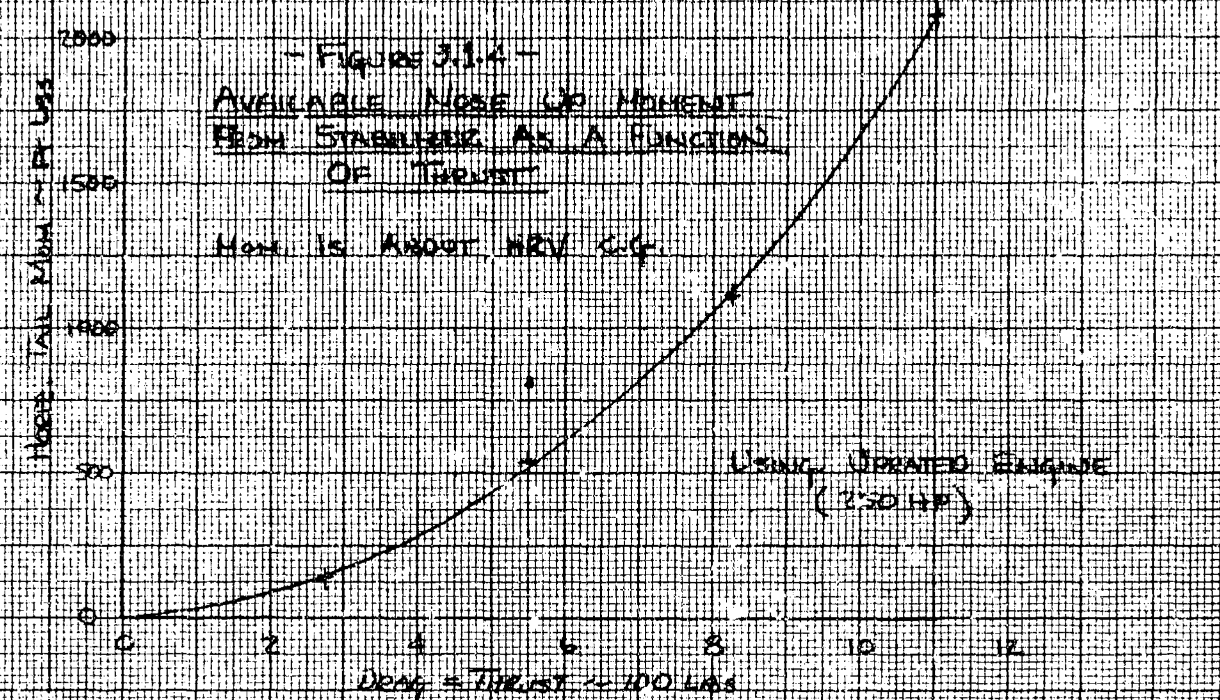


FIGURE 3.1.3- DRAG VERSUS SPEED FOR 2:1 ELLIPTICAL SECTION (SINGLE FLOAT)



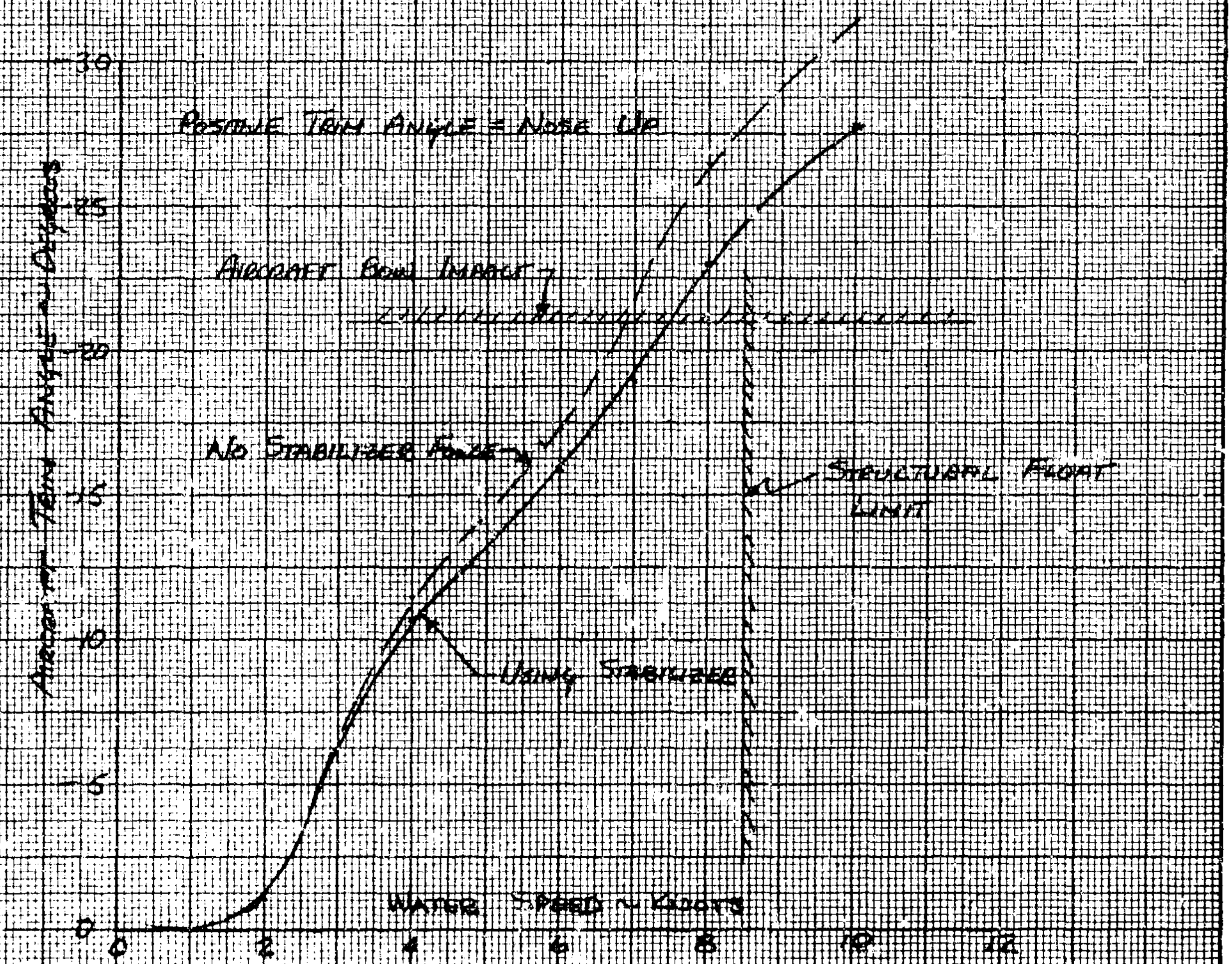


FIGURE 3.1.5 – HRV TRIM ANGLE VERSUS WATER SPEED

The preliminary float sizing method was similar to the one used in Reference 1. Basically, this derivation of float size determination was based on an empirical approach derived from the Navy's requirement in SR 59B. This specification defines the required displacement of tip floats. Relating this equation to a Vertical Float System yields the following equation:

$$A_R = \frac{W}{2\rho \ell^2 \sin\theta} \left[h \sin\theta + \frac{0.1\ b}{W/S} + .06\sqrt[3]{W}\right]$$

where:

A_R = Required float cross section (ft^2)

W = Weight of the aircraft

ℓ = Float span

ρ = Density of sea water (64 lb/cu ft)

θ = Angle of heel desired

S = Wing area (ft^2)

h = Effective height of cg above center of buoyancy of float (ft)

b = Span

The first term of this equation represents a neutral stability requirement, while the other terms define a wind under the wing and righting moment margin. In using this equation for the pitch plane, the wind under the wing term was dropped since it does not reflect a realistic condition.

The basic design data and configuration for the HRV airplane are shown in Figure 3.2.1. Using this data, the required float diameter was determined for h varying from 4 to 12 feet and float span varying from 3 to 20 feet. The results are shown in Figures 3.2.2 through 3.2.4. Figure 3.2.2 shows the required float diameter to maintain neutral stability for a 14° heel angle. Figures 3.2.3 and 3.2.4 show the required float area in the pitch and roll planes as per SR 59B, respectively.

The selected float dimensions were now evaluated as per the aircraft configuration. Keeping in mind that minimum airplane modification work be done and keeping the amphibian qualities, it was decided to use four hull floats (two forward and two aft) and two wing floats. Keeping all the float dimensions equal to minimize float construction cost, the pitch plane float selection was critical. Using this, the following float dimensions were selected:

Hull Floats (2) - Area = 3.8 ft^2 Length = 4.72 ft.

Wing Floats - Area = 1.9 ft^2 Length = 2.42 ft.

This results in a buoyant force distribution of 79.5% of the weight on the hull floats and 20.5% on the wing floats. In order to minimize the drag for the water maneuver requirement, an elliptical cross sectional shape with a 2:1 chord to thickness ratio was used. This results in a float cross section which is 2.2 foot long and 1.1 foot wide. The general arrangement of the HRV aircraft with vertical floats is shown on drawing 818171. The resulting float span for the wing was 24 feet while the hull float longitudinal span was 14.6 feet. The wing and forward hull float span was predicated by aircraft structural provisions. Float centerline locations were kept as symmetrical as possible with the existing HRV cg location in order to minimize any fore-aft shift.

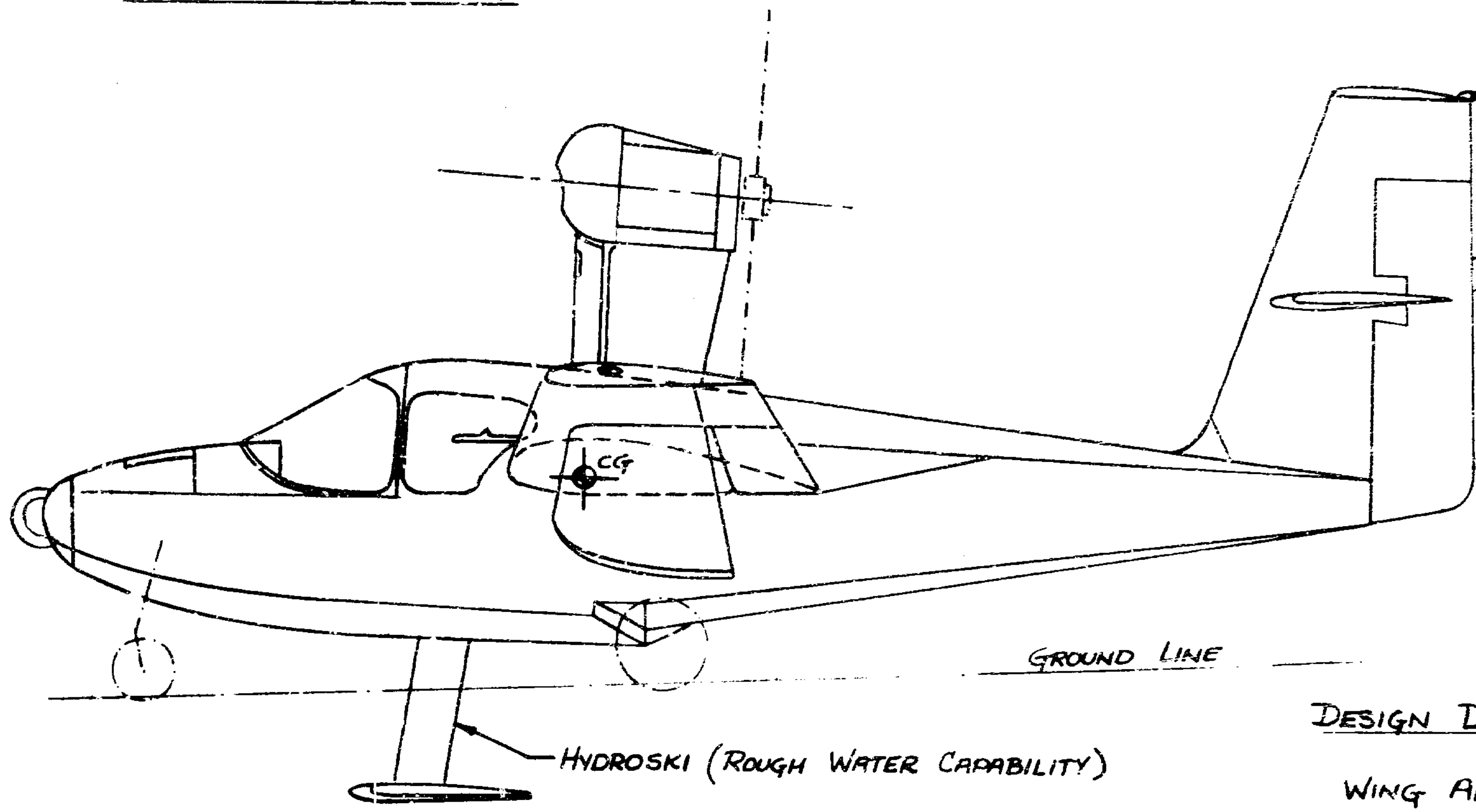
Using these values, a summary of the HRV Vertical Float System practical dimensions follows:

	Hull Float	Wing Float
Load Distribution	79.5%	20.5%
Displacement - lb	1193 (2)	307
Float Dimension (Ellip. Axis)	2.2' x 1.1'	2.2' x 1.1'
Float Area	3.8 ft^2 (2)	1.9 ft^2
Submerged Length	4.72 ft.	2.42 ft.
Total Length	7.7 ft.	7.7 ft.
Semi Span (from cg)	7.3 ft.	12 ft.

Initially, it was hoped that the HRV airplane with the Vertical Float System would represent a scale version of the P5A airplane as reported in Reference 1. The major difference occurs in the percentage of weight displaced by the hull and wing floats. The P5A uses a 70%-30% split between hull and wing while the HRV is close to a 80%-20% ratio. A study was made to determine sensitivity of the support ratio to float cross sectional area for the HRV airplane. The results are shown in Figure 3.2.5. It becomes apparent that for the HRV airplane it would be impractical to use the 70%-30% ratio used in the P5A airplane. The only way to change the buoyant displacement ratio is to increase the float spans. This would mean extensive additional rework for the HRV airplane which from the study ground rules is undesirable.

The resulting final selected float system configuration will give a reserve buoyancy of 22% in pitch and 67% in roll. These are considerably higher than those of the P5A aircraft. With the associated downward shift of the aircraft cg due to the Vertical Float System weight, the reserve righting moment margin will increase still more thus providing ample stability. This is a desirable direction in any test type program which seeks to prove a design concept.

HRV - CONFIGURATION



DESIGN DATA:

WING AREA	170 $FT.^2$
SPAN	38 FT
LENGTH	23.5 FT.
FLOAT SPAN	24.0 FT.
WEIGHT	2900. LBS

FIGURE 3.2.1 - HRV GENERAL CONFIGURATION

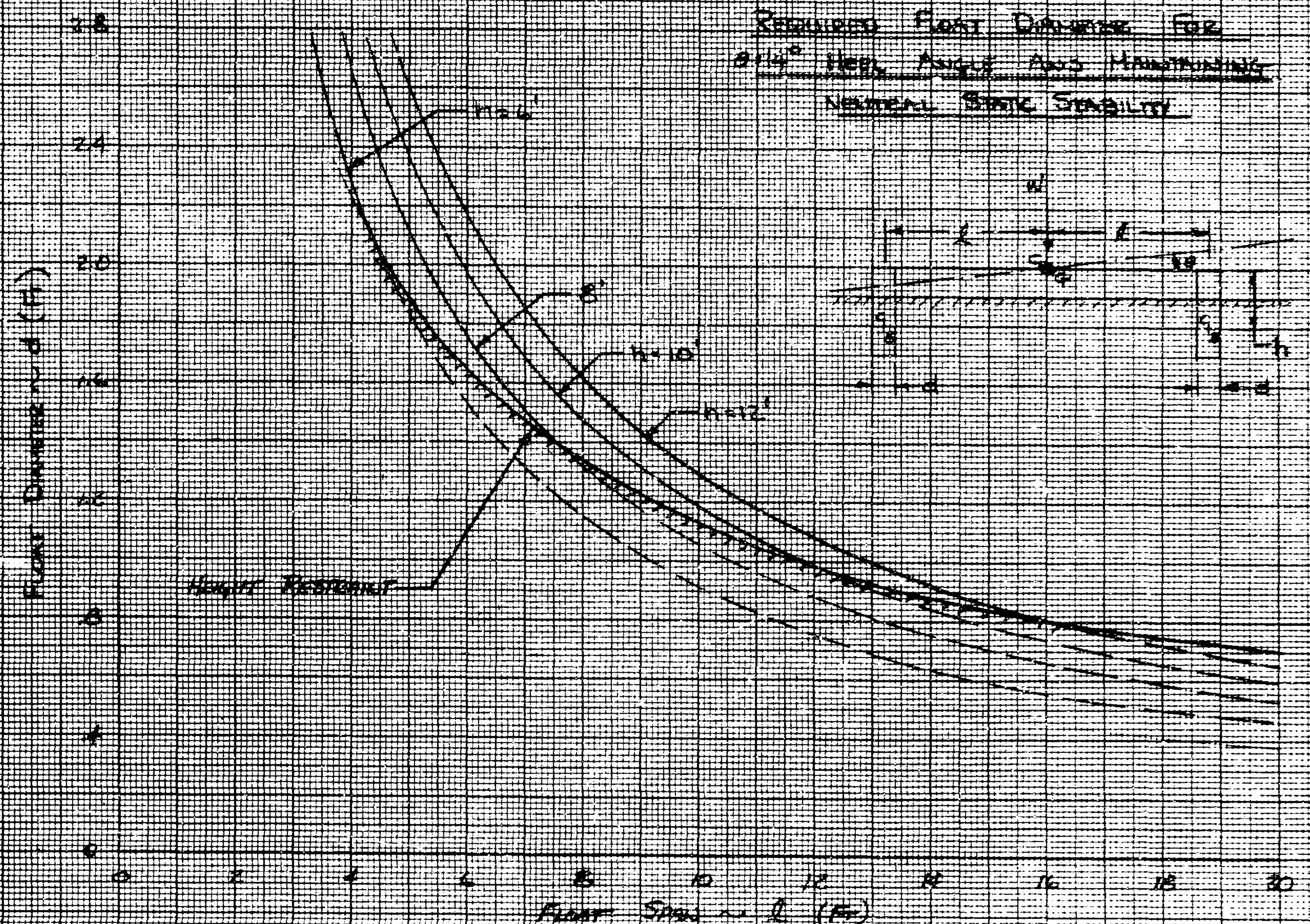


FIGURE 3.2.2 – FLOAT DIAMETER REQUIRED FOR NEUTRAL STABILITY

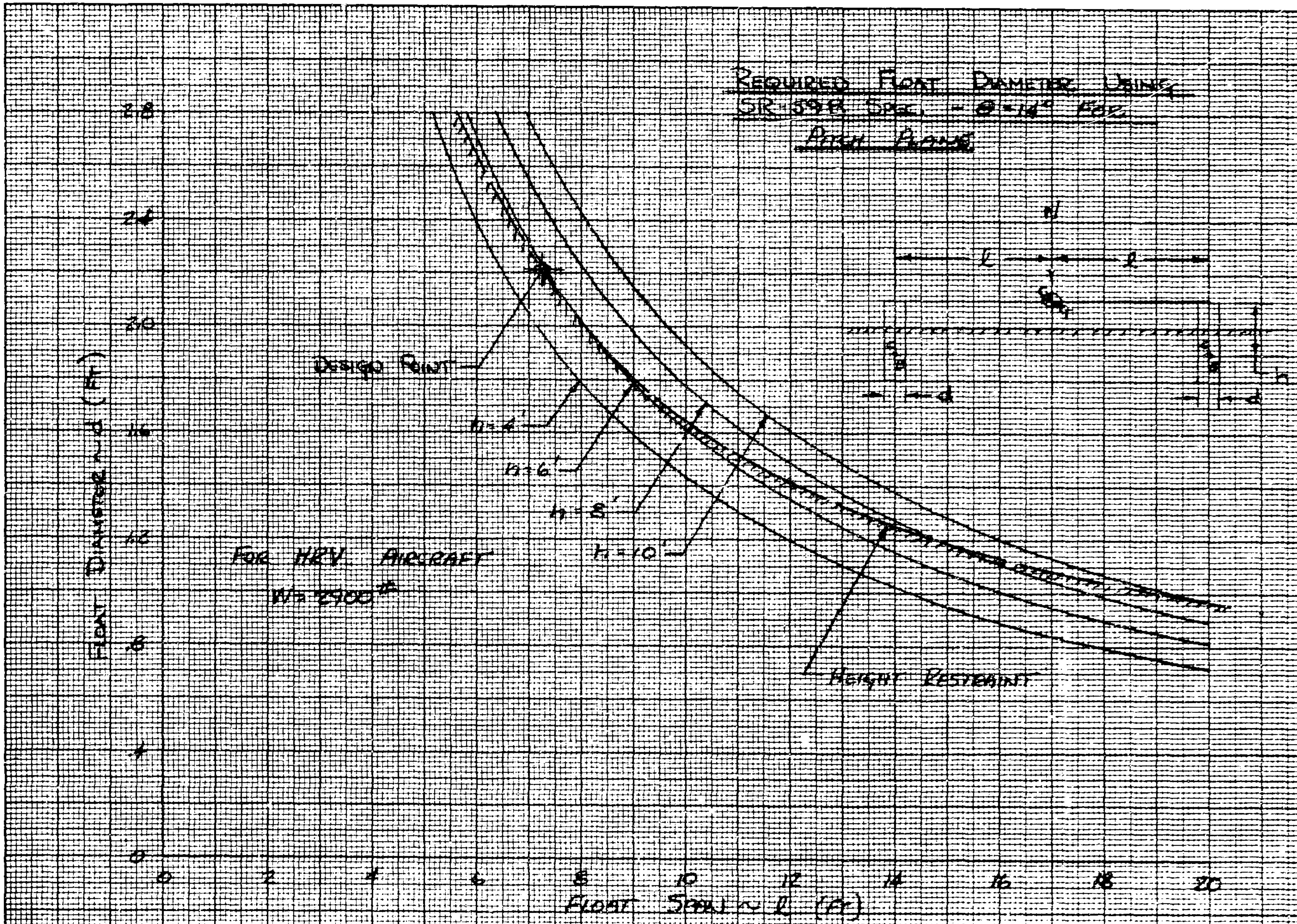


FIGURE 3.2.3 - FLOAT DIAMETER REQUIRED IN PITCH PLANE USING SR-59B

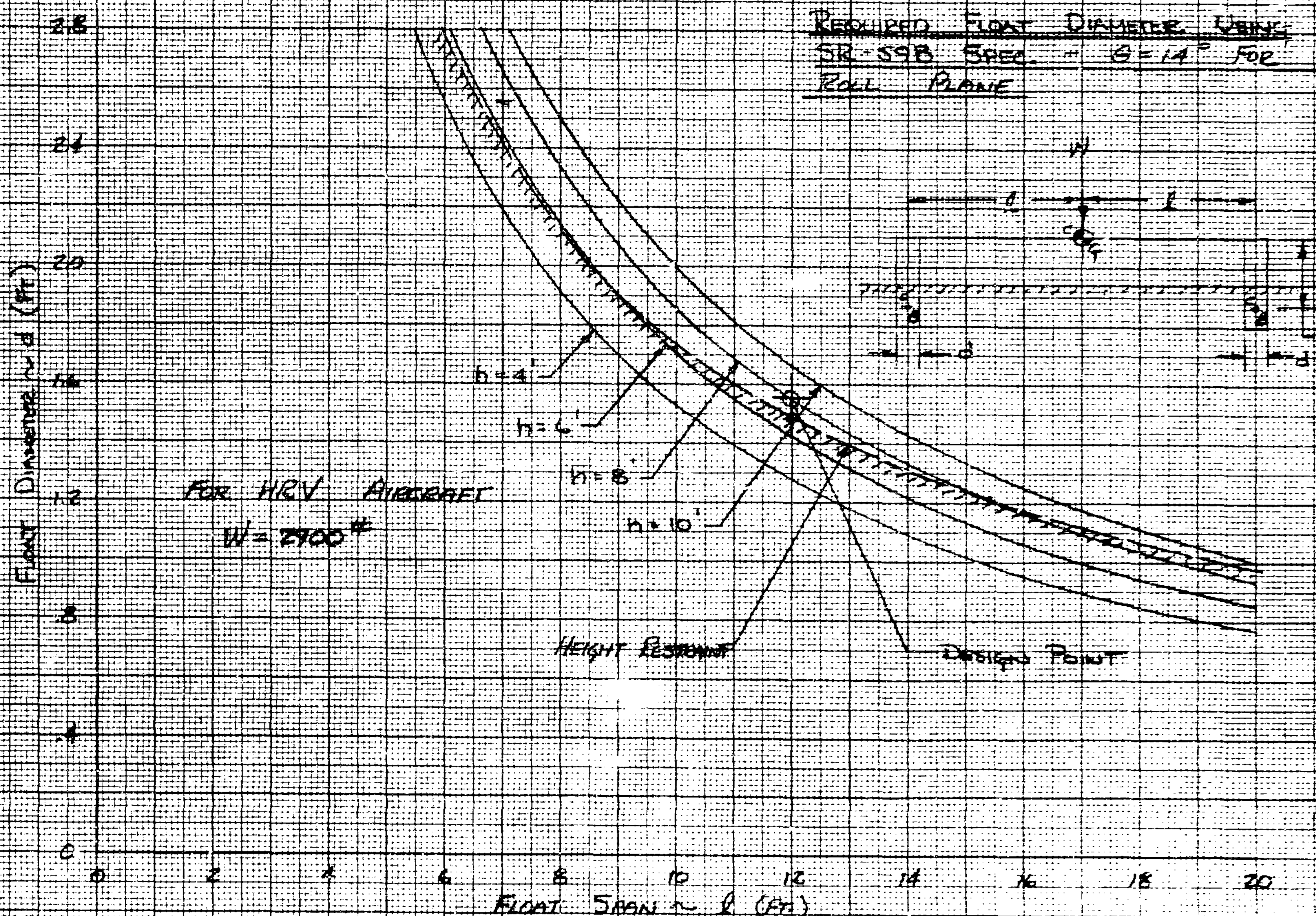


FIGURE 3.2.4 - FLOAT DIAMETER REQUIRED IN ROLL PLANE USING SR-59B

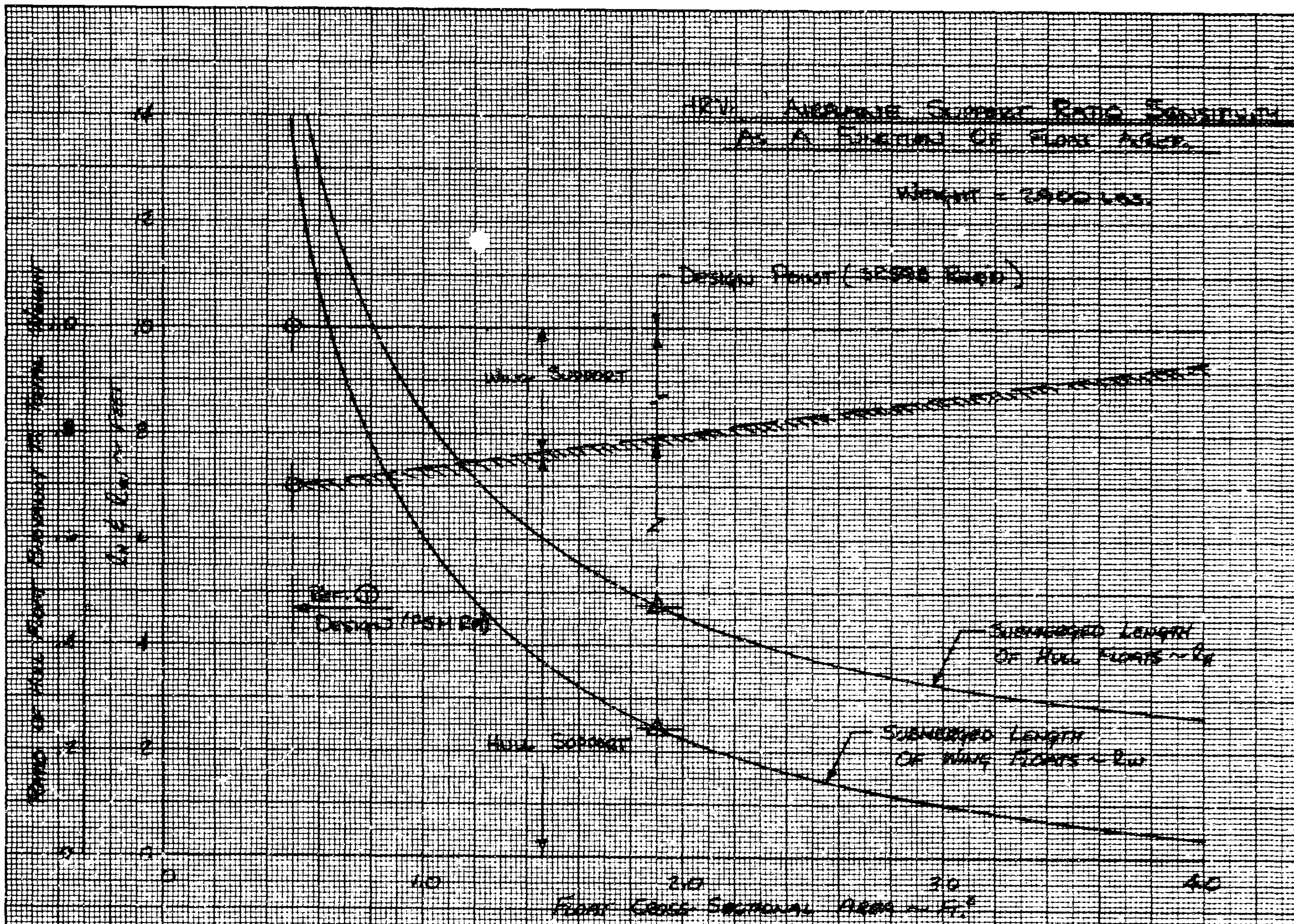


FIGURE 3.2.5 - WING AND HULL FLOAT LENGTH AND BUOYANCY - WEIGHT RATIO

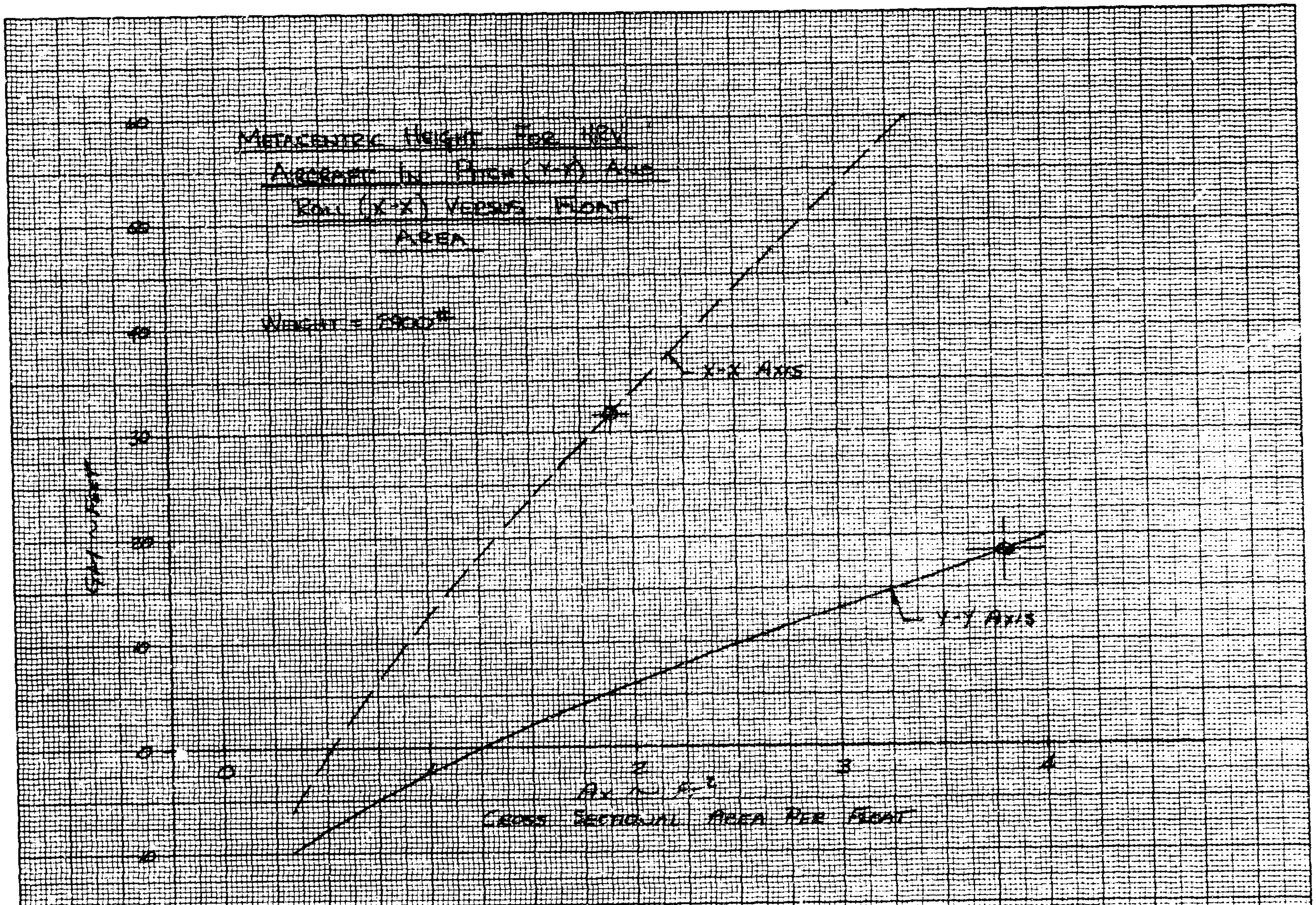


FIGURE 3.2.6 – HRV METACENTRIC HEIGHT AS A FUNCTION OF FLOAT CROSS SECTIONAL AREA

A final check on the float system was made by determining the metecentric height. This is summarized in Figure 3.2.6. For both the roll and pitch axis, the metecentric height is positive.

3.3 Hydrodynamic Considerations

A great number of studies have been made evaluating the response characteristics of various vehicles and float configurations in a seaway (References 3 through 8). Since the primary goal of this study was to prove the feasibility of the Martin Marietta Vertical Float design concept, no formal hydrodynamic studies were made. Additionally, the demonstration HRV airplane represents a 1/3 scale model of the P5A aircraft. The P5A platform has had extensive model hydrodynamic testing (References 1, 10 and 11). All of these indicated desirable hydrodynamic features.

The major conclusion of all the studies and test results indicate that the Vertical Float Systems must have damping devices incorporated to minimize response. The studies conducted in both References 1 and 9 used a damping plate area to float cross section area ratio of 3.0. The investigation of Reference 5 used a ratio of 5.0. The dampers which are recommended for the HRV aircraft Vertical Float System, used a ratio of 3.0. Considering the construction of the actual float bag, additional inherent damping characteristics exist. The elliptical cross section has a greater wetted area than a circular one; the extension of the ribs beyond the outer perimeter of the inflated shape will act as damping devices, and the torus type inflated segments will impede continuous vertical flow. As a result, it is felt that the scale test of the demonstration HRV will have much more damping than its prototype P5A.

3.4 Detail Design

This section discusses the studies made and results to prove the feasibility of the Inflatable Vertical Float System. The studies made did not seek to optimize the total system weight, but rather to prove the resulting system was workable within the established design weight goal. For optimum airborne hardware, a price per pound of weight may be evaluated and worthwhile savings might be realized. There are four basic components which comprise the proposed Inflatable Vertical Float design concept. These are the main telescoping support tube, the inflatable bag and metal ribs, the extension/retraction mechanisms and the inflation system. The following sections discuss these components in detail.

3.4.1 Telescoping Support Tube

A design evaluation of the telescoping support tube dictated a full understanding of the complete vertical float load system. Three essential structural elements make up the load transfer system. These are the inflated bag, the support ribs, and the main tube. Figure 3.4.1-1 schematically indicates how the applied loads are reacted through the structural system. Part (a) of 3.4.1-1 shows the inflated bag. Since

the maximum water head pressure is 3.42 psi and the dynamic pressure due to a 6-knot velocity is 1.40 psi (totaling 4.82 psi), an internal working pressure of 5 psi was selected. Thus, the bag fabric is kept always in tension and the support column locked in the extended position.

As is indicated in part (b) of Figure 3.4.1-1, the ribs serve two essential functions. First, they provide the lateral strength to the bag (maintain the cross sectional shape) and second, they transfer axial and torsion loads to the support column. The two end ribs, in addition to performing rib functions, also act as pressure bulkheads and transmit the net pressure force to the support column. Then, as shown in part (c) of Figure 3.4.1-1, the support column carries the net forces to the aircraft.

A survey of the float load conditions showed that the most critical condition for structural design was the 6-knot condition. This condition results in both maximum lateral and moment loadings for the support tube. The relative attitudes of the float structure during load conditions are shown in Figure 3.4.1-2. Aircraft drift conditions with the associated wave surface load pertubations were also investigated but did not exceed the load environment of the 6-knot condition.

Taking the critical loading condition, an iterative type solution was applied in order to determine the final tube moments. This is a typical approach to laterally loaded column structure. The resulting support tube shear and moment curves are shown in Figure 3.4.1-3.

The structural design problem involved in a telescoping load support member is to provide adequate strength and stability and yet have structural elements which will not bind under load and deformations. The design philosophy used is essentially as follows. While the support tube is in the extended position, it must have maximum load carrying capability. Therefore, in this position, the joint configuration tolerance is held to a minimum. Since during retraction the critical design loads are significantly less, the tube tolerances may be increased to provide greater freedom of motion.

The determination of required tube and joint thicknesses involved the evaluation of numerous design parameters. The preliminary tube configuration is shown in Figure 3.4.1-4 along with a typical joint cross section. Taking the critical applied loads from Figure 3.4.1-3, the tube thickness was determined for overall strength requirements. Using these thicknesses, a step column analysis was performed to determine the allowable column load. The results indicated that overall column stability was not critical.

As far as local tube stability is concerned, the largest D/t value is 21.3 which is again not a structural design restraint. The structural design of the joint to transfer moment was accomplished by short coupling the moment over 2/3 the joint length. This assumes that the pressure distribution is triangular, with a peak at the joint-tube

ends. The equivalent triangular load was then applied to the tube as both a point load and a circumferentially distributed load varying as a cosine function over 90°. A visual representation of this may be seen in the sketch on Figure 3.4.1-5. Generally, the thickness required equation takes the following form (References 11-13).

$$t = c_T \frac{P}{\sigma}^{1/2}$$

where

P = Equivalent load (lb)

σ = Allowable stress (lb/in^2)

c_T = Factor reflecting load take-out assumption

t = Thickness (in)

The variation of c is shown in Figure 3.4.1-5 for a concentrated load case and a distributed load case. From this figure, it is obvious that for the concentrated load case, the required thickness is a function of effective width only while in the distributed case, the thickness required is a function of both effective width and radius.

A typical variation of thickness required as a function of moment, tube overlap, and tube diameter is shown in Figure 3.4.1-6. This figure is based on the concentrated load case. In reality, this approach tends to predict conservative structural requirements since there is no path for a concentrated load input. Actually, the joint design problem is one of compatibility between structural and mechanical deflection. Since it was felt that a formal theoretical investigation could not be justified during this study and in order to gain insight into the actual hardware problems which might exist, a sample joint was made at company expense. Figures 3.4.1-8 and 3.4.1-9 show a picture of the model joint.

A graphical representation of the selected tube and joint thicknesses as compared to the required thickness is shown in Figure 3.4.1-7. Generally, it shows that the lower tube section is critical for overall bending, and the joint thickness is determined by the uppermost joint bending requirements. In order to minimize the production costs, a constant thickness tube was selected along with a constant thickness joint. The tube diameter variation is a function of joint thickness. It appears apparent that significant weight reduction could be accomplished; however, the requirement of the tubes sliding within one another dictates that the only way in which additional thickness could be removed is by machining or chem-milling pockets into the tubes to approximate the tapering design requirements. This is too costly and is not recommended.

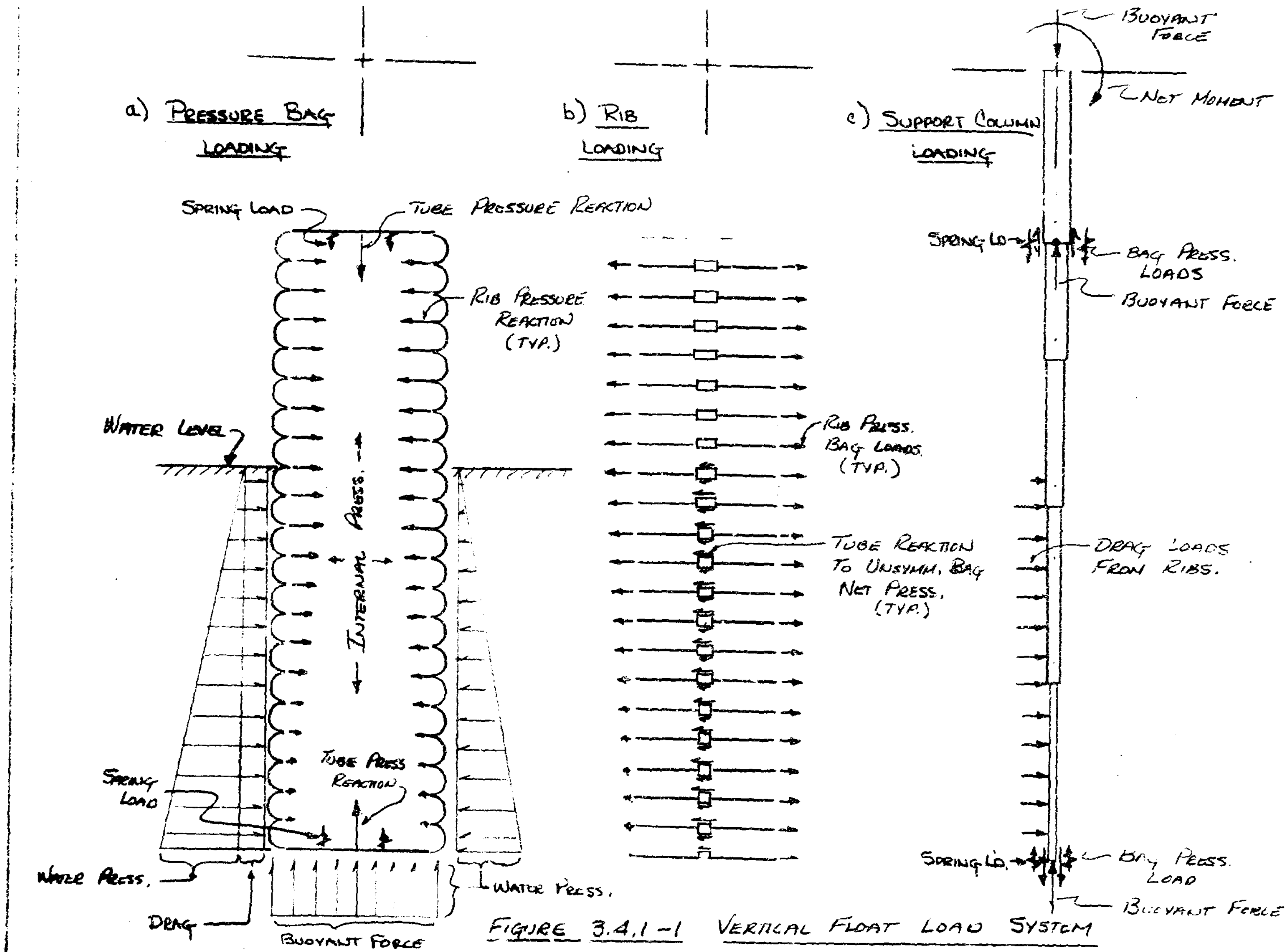


FIGURE 3.4.1-1 VERTICAL FLOAT LOAD SYSTEM

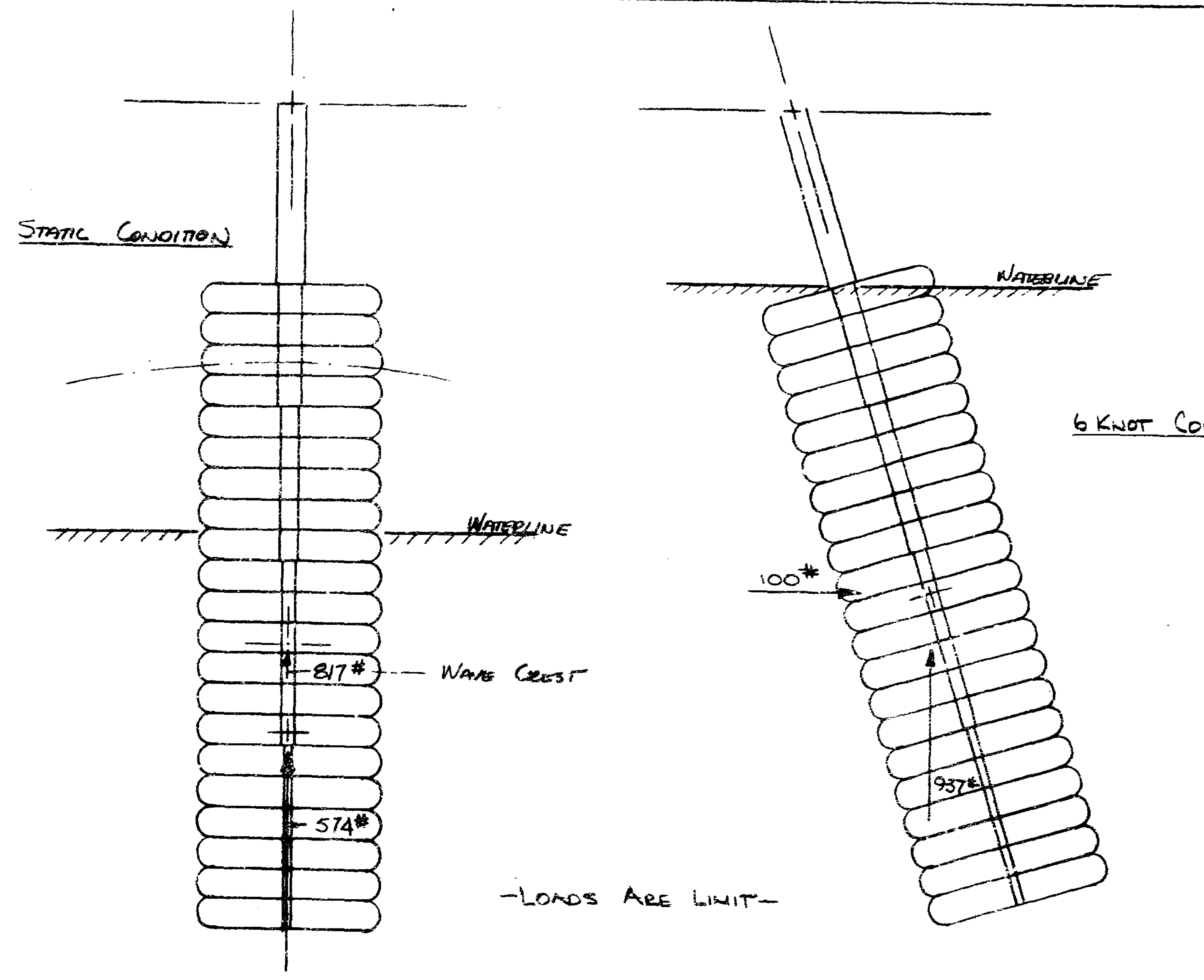


FIGURE 3.4.1-2 - VERTICAL FLOAT DESIGN CONDITIONS

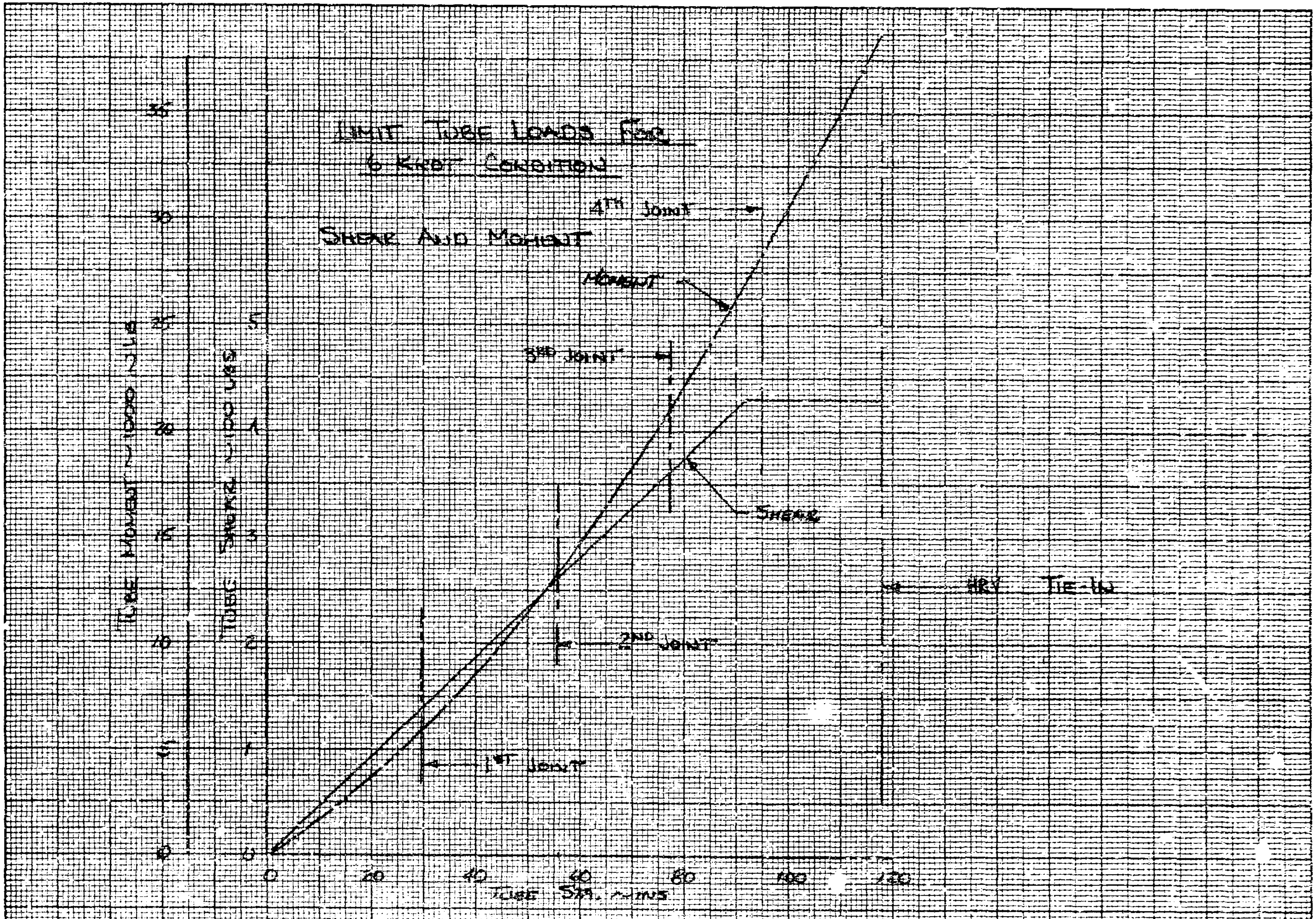
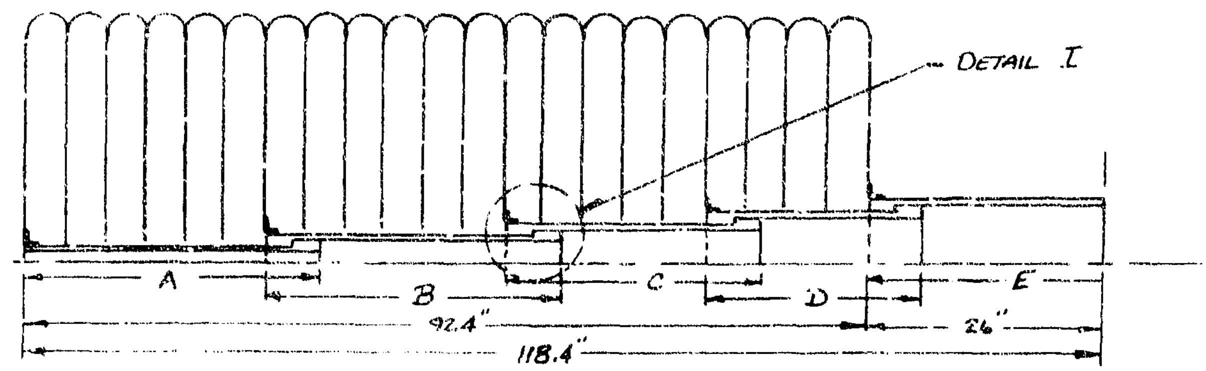


FIGURE 3.4.1-3 - COMMON TUBE LOADS



— TUBE DIMENSIONS —

TUBE SECTION	OUTSIDE DIAMETER	INSIDE DIAMETER	LENGTH
A	1.0 IN.	.531 IN.	32.4 IN.
B	1.75 IN.	1.375 IN	32.4 IN.
C	2.50 IN.	2.125 IN	28.0 IN.
D	3.25 IN.	2.875 IN	23.6 IN.
E	4.00 IN.	3.625 IN.	26.0 IN.

DETAIL I

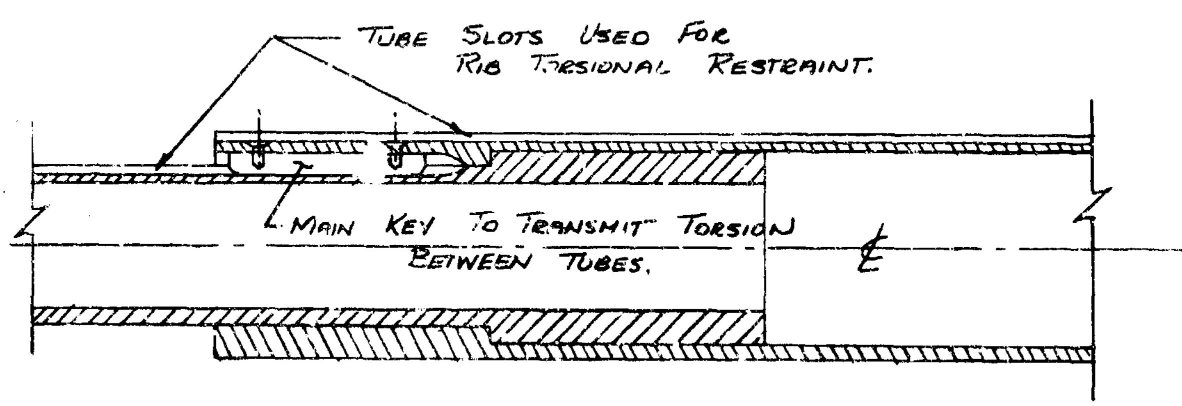


FIGURE 3.4.1-4 - PRELIMINARY SUPPORT TUBE CONFIGURATION

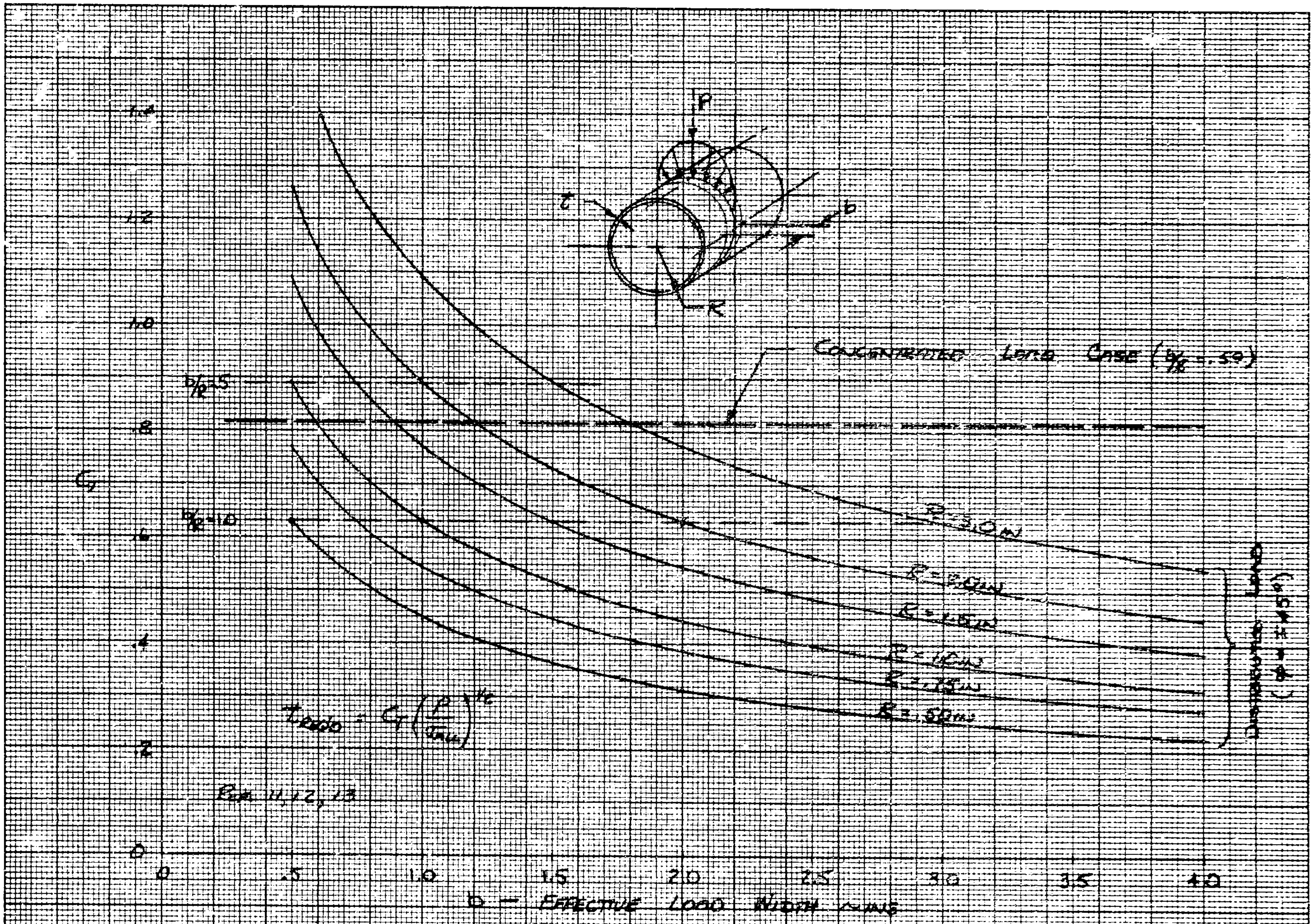


Figure 3.4.1-5 — Tube Thickness Factor (Vibration)

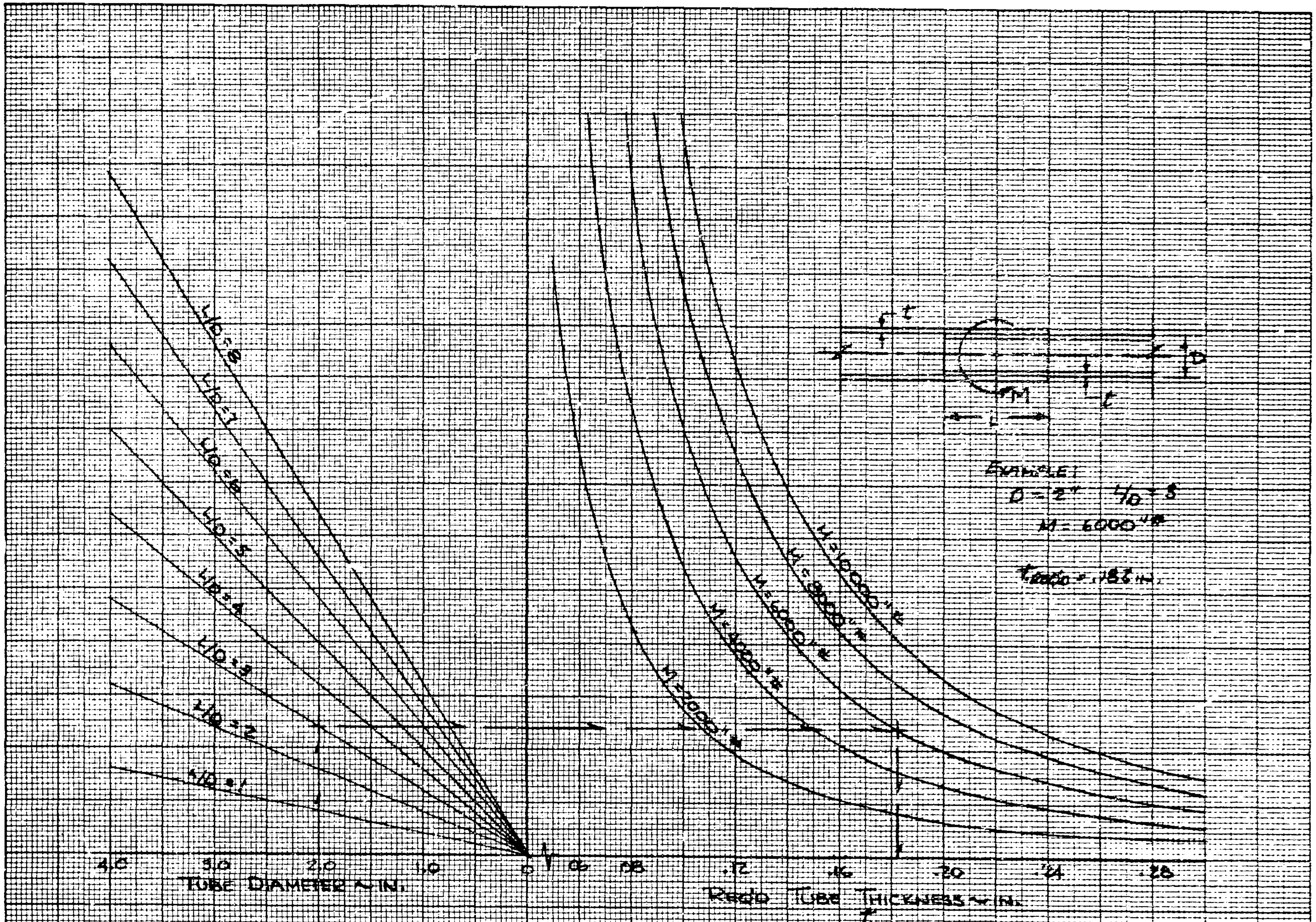


FIGURE 5.4.1-6 - JOINT DESIGN NOMOGRAM

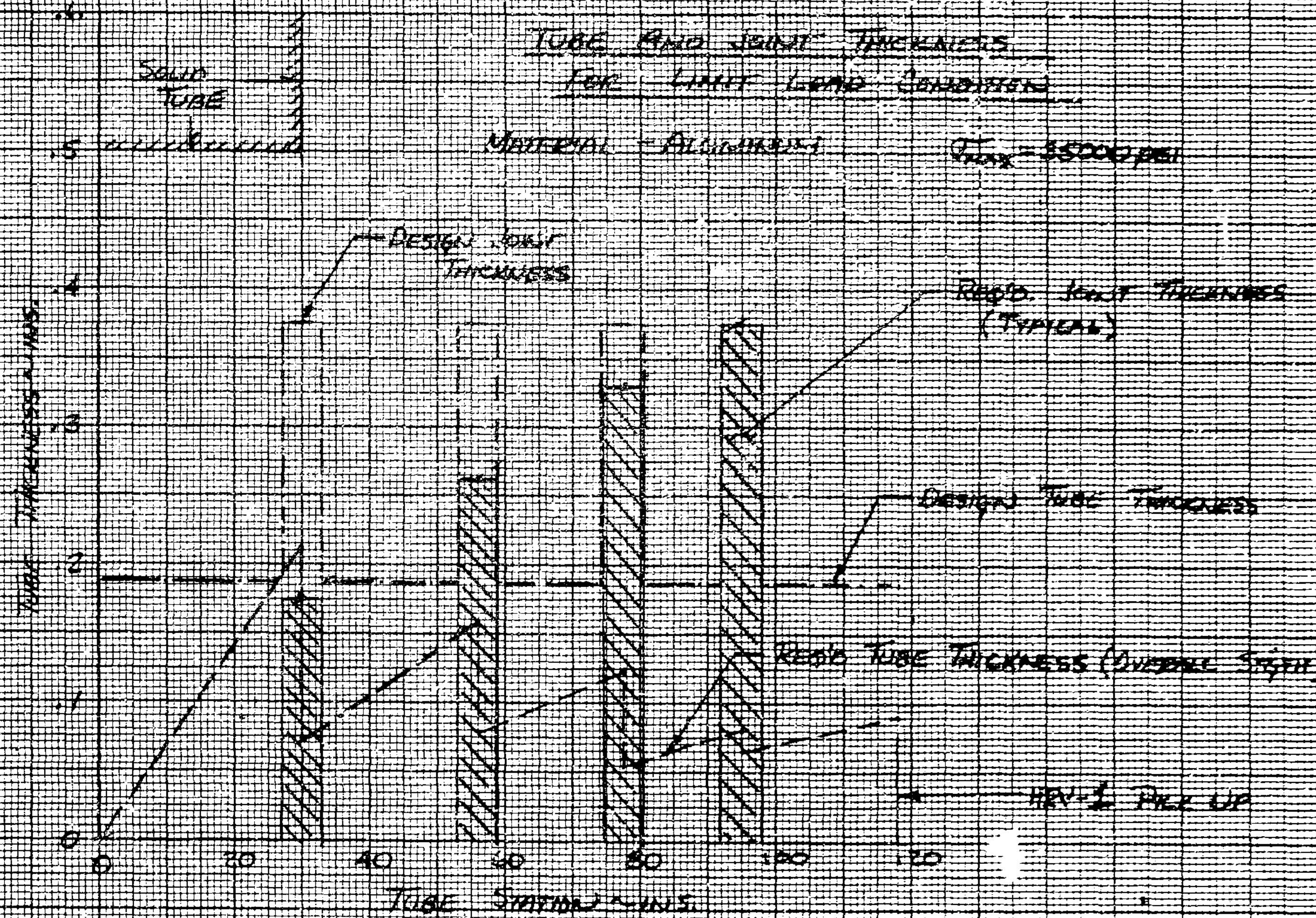


FIGURE 3.4.1 - 7 - TUBE THICKNESS RELATIONSHIP

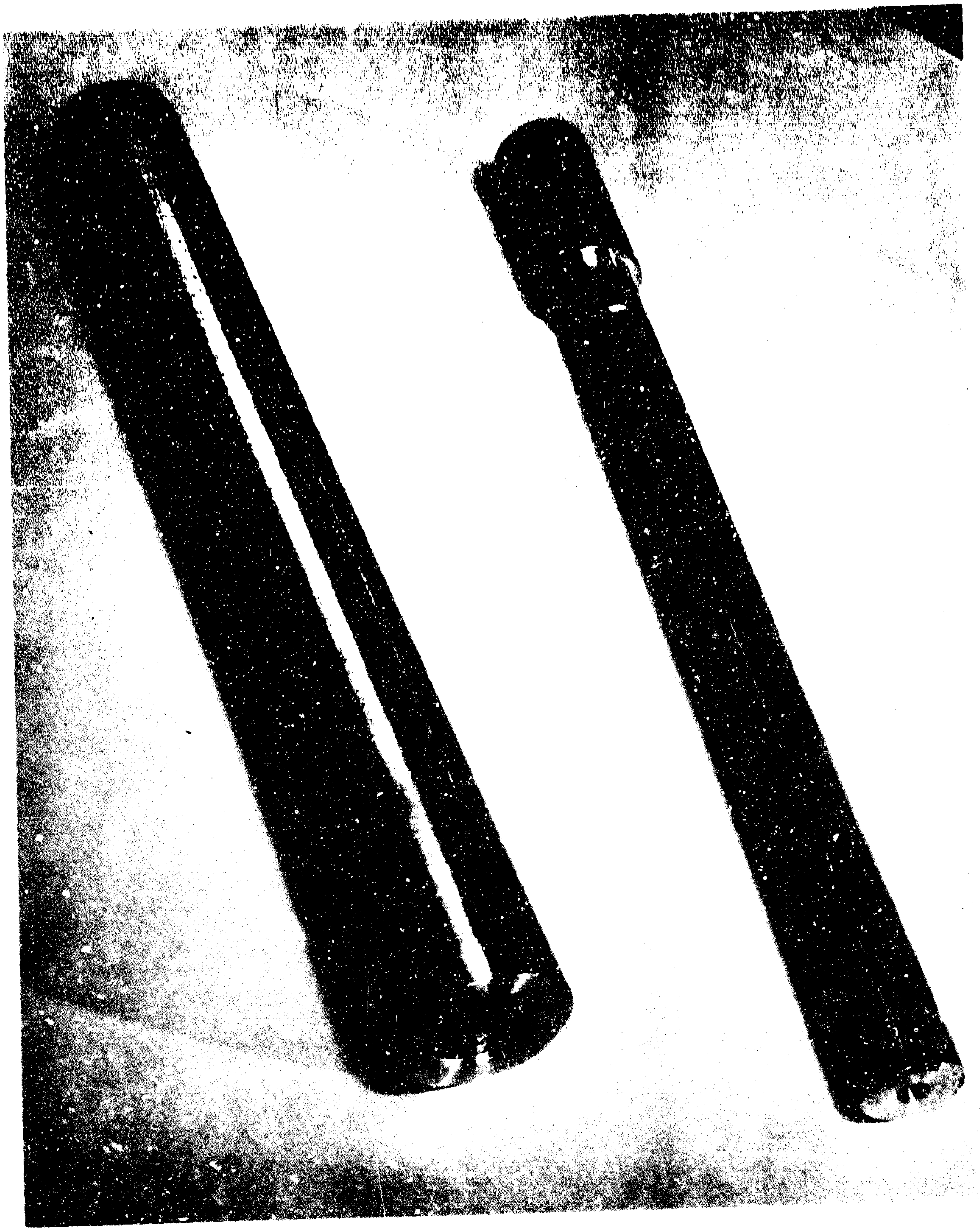


FIGURE 3.4.1.8 TUBE JOINT

The elliptical cross section which is used for the float is doubly symmetric which under zero degree angle of attack will have its load center at the 50% chord point with no resulting torque. Under angle of attack conditions, however, the load center could move slightly forward (40%) and result in torque loads. These torque loads are transmitted through the tube sections by virtue of the joint key. The individual ribs transmit local torque to the tube through the full length tube slot and rib finger.

The ribs were sized by considering them as elliptical frames loaded periferially and reacting loads at the tube. Only the ribs which are located at tube junctions are fixed to the tube in all directions. The intermediate ribs are "floating" by virtue of the rib tube and rib finger-rib slot tolerances. Under load, the ribs will "bottom" out and transmit load into the tube.

The entire structural support system analysis considered that the inflatable bag would take no applied loads. Actually, the bag will have inherent bending and torsional stiffness. The load ability will nevertheless represent a second order effect.

3.4.2 Flotation Bag Test Unit

The design concept developed for construction of the flotation module assumes that the module assembly will be restricted in vertical movement, so that the aluminum formers will be constrained from exceeding the maximum used in design of the module.

In the current design for the flotation module, this maximum separation was established at 4.0 inches. The flexible portion of the module was then designed to conform to a radius of 2.0 inches when fully inflated.

Transfer of the stress from the flexible inflated fabric to the rigid aluminum formers is accomplished through an adhesive bond developed between the fabric and the aluminum former.

Design Parameters

1. Operating Pressure, 10 psi

2. Design Pressure, 50 psi (min.)

3. Former Spacing, 4.0 inches

4. Module Radius, 2.0 inches

Stress Analysis

The applied stress in a typical section taken through the flexible module wall is shown in Figure 3.4.2.1.

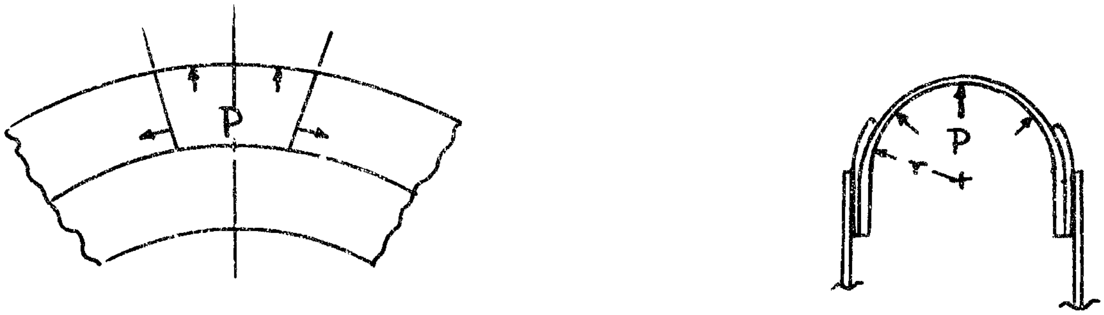


Figure 3.4.2.1

A practical approximation of the wall stresses developed during pressurization is determined as follows:

$$s_g = Pr = 50(2) = 100 \text{ lbs/inch}$$

$$e_g = \frac{s_g}{E} = \frac{100}{512} \times 100 = 19.5\%$$

$$e_a = \frac{e_g r}{C_o} = \frac{0.195(2.0)}{70} \times 100 = 0.56\%$$

$$s_a = e_a E = 0.0056(512) = 2.9 \text{ lbs/inch}$$

where:

s_g = Circumferential stress, lbs/inch

P = Pressure psi

r = Radius, inches

e_g = Circumferential elongation in wall, percent

E = Modulus of coated fabric

e_a = Axial elongation in wall, percent

C_o = External perimeter of module wall, inches

s_a = Average axial stress, lbs/inch

The results of this approximation show that the major stress occurs in the circumferential or girth direction. Therefore, the design parameters to be met in the flexible wall structure follow.

1. Minimum wall girth strength in 45° filament bias fabric - 100 lbs/inch

2. Minimum shear adhesion of 45° filament bias fabric to aluminum former - 100 lbs/inch

This adhesion level is required in order to permit developing the 100 lbs/inch stress in the flexible wall.

Construction

Fabric. Uniroyal coated fabric 5200 was chosen as the material to be used in the construction of the four module flotation unit shown in Figure 3.4.2.2. This is a 2 oz nylon fabric with equal strength in both warp and fill, coated with a NBR elastomer. The coated weight is 0.045 lbs/sq ft. This coated fabric is essentially leak tight; however, in order to avoid the possibility of developing or incorporating a pin hole defect, it is used as a two-ply construction. This two-ply 45° bias construction has a demonstrated minimum strength of 128 lbs/inch of width and, therefore, meets the minimum fabric strength requirement.

Adhesion System. A critical parameter for the successful performance of the inflated flotation module is maintaining adequate bonding of the 5200 fabric material with the restraining aluminum former. A Uniroyal proprietary metal preparation process was used to prepare the Alclad aluminum series proposed for construction of the formers which included the following:

Alclad - 2024-0
Alclad - 6061-T6
Alclad - 2024-T3

The prepared metals were then treated for adhesion using a Uniroyal proprietary system involving the use of Ty Ply BN. Referring to Figure 3.4.2.2, the 0.010" layer of 3010 NBR elastomer was then bonded to the aluminum former, using heat and pressure. Each former was visually inspected for continuity of adhesion. Test samples duplicating this adhesion system indicated a shear strength of 194 lbs/inch width of seam when the sample had a 1.0-inch minimum adhered length. This system, therefore, met design requirements.

Adhesion of the 5200 coated fabric to the bonded 3010 was achieved using Uniroyal adhesive 3242. Test samples prepared with this adhesive system showed a shear strength of 104 lbs per inch width when the sample had a minimum adhered length of 1.0 inch. This shear strength would be increased approximately linearily up to 1.5 inches. This adhesive system also met minimum requirements.

Fabrication of Test Modules

Single Module Test Unit. A single test unit was prepared to verify the construction to be used in fabricating the four module test unit. Figures 3.4.2.5 and 3.4.2.6 show the overall single cell unit. The fabrication process discussed for the four module unit was followed in its entirety.

The primary objectives for constructing this single unit covered the following:

a. Verify inflated shape of module.
b. Verify minimum burst strength.
c. Observe collapsability of module.

Pressurization Test - Single Module. The module was inflated with air to 5 psi and the surface soap tested. No leaks were found. The module shape conformed satisfactorily to the anticipated design shape.

The test unit was then collapsed to observe ability to nest. A very normal pattern of collapse was observed indicating no design change required to favor module nesting during collapse.

The test unit was then held at 10 psi for 24 hours with no observed change in module properties.

In a final pressurization to failure test, the module was pressurized with water to 60 psi, at which time the restraining test jig failed and the test was terminated. However, the module had not failed at this pressure. Figure 3.4.2.5 shows the overall single module test specimen.

The performance of the single module test unit confirmed the adequacy of the design proposed for the four module test unit.

Fabrication of the Four Module Test Unit

The four module test unit is shown in general design in Figure 3.4.2.2. Details of construction are shown in Figure 3.4.2.3 and Figure 3.4.2.4.

The detail of preparing the aluminum former for attachment to the flexible wall structure is shown in Figure 3.4.2.3. The fabrication technique used was discussed under the section on adhesion and will not be repeated here. An important attribute of this method of construction

is that each former may be critically examined for quality of adhesive bond to the former, with the option of rejection of defective parts. This insures that there will be no exposure to adhesion failure in a full scale model involving 20 to 30 assembled modules.

The prepared aluminum formers are then adhered to the flexible wall structure shown in detail in Figure 3.4.2.4.

The basic module is constructed on a soak out mandrel conforming with the inflated shape of the module shown as (1). The 5200 coated fabric material is then fabricated into the wall structure as indicated in Figure 3.4.2.4. Adhesive 3200 is used to adhere the various fabric layers. A 0.010" NBR gum tape 5137 is used to seal the respective edges of the 5200 fabric shown as (6).

Review of Figure 3.4.2.4 shows that four layers of 5200 fabric are involved at the aluminum former extending out approximately 1.5 to 2.0" from the former. This added thickness should serve to minimize possible damage to the fabric by the metal former. The two ply of fabric shown at the outer periphery of the module has adequate strength, but because of its thinner wall section serves to facilitate "cupping" of the module for good nesting during collapse for final packaging. Figures 3.4.2.7 and 3.4.2.8 show the final fabricated four module units.

Proof Testing of Four Module Test Unit

The four module test unit was pressurized to 5 psi with air and soap tested. No leaks were observed.

The test unit was then inflated with air to 10 psi for 24 hours. No structural change was observed in the test module upon completion of this test. The test unit was collapsed for packaging. The collapsed configuration is shown in Figure 3.4.2.9. Measurements made on the collapsed unit were as follows:

(a) Thickness of four collapsed modules through the metal formers including the two terminal formers each 0.25" thick was 1.5 inches.

(b) Thickness of the four collapsed modules at outer periphery of flexible wall 3.1 inches. This thickness dimension includes the cupping of the first module which in itself was 2.0 inches.

General Comment

The four module test unit was fabricated of NBR coated 2 oz nylon fabric. The NBR fabric was used as an expedient because it was available as a production item at Uniroyal. This material is functionally satisfactory for test evaluation purposes. The NBR coating is deficient in outdoor aging properties when compared with the preferred Neoprene or

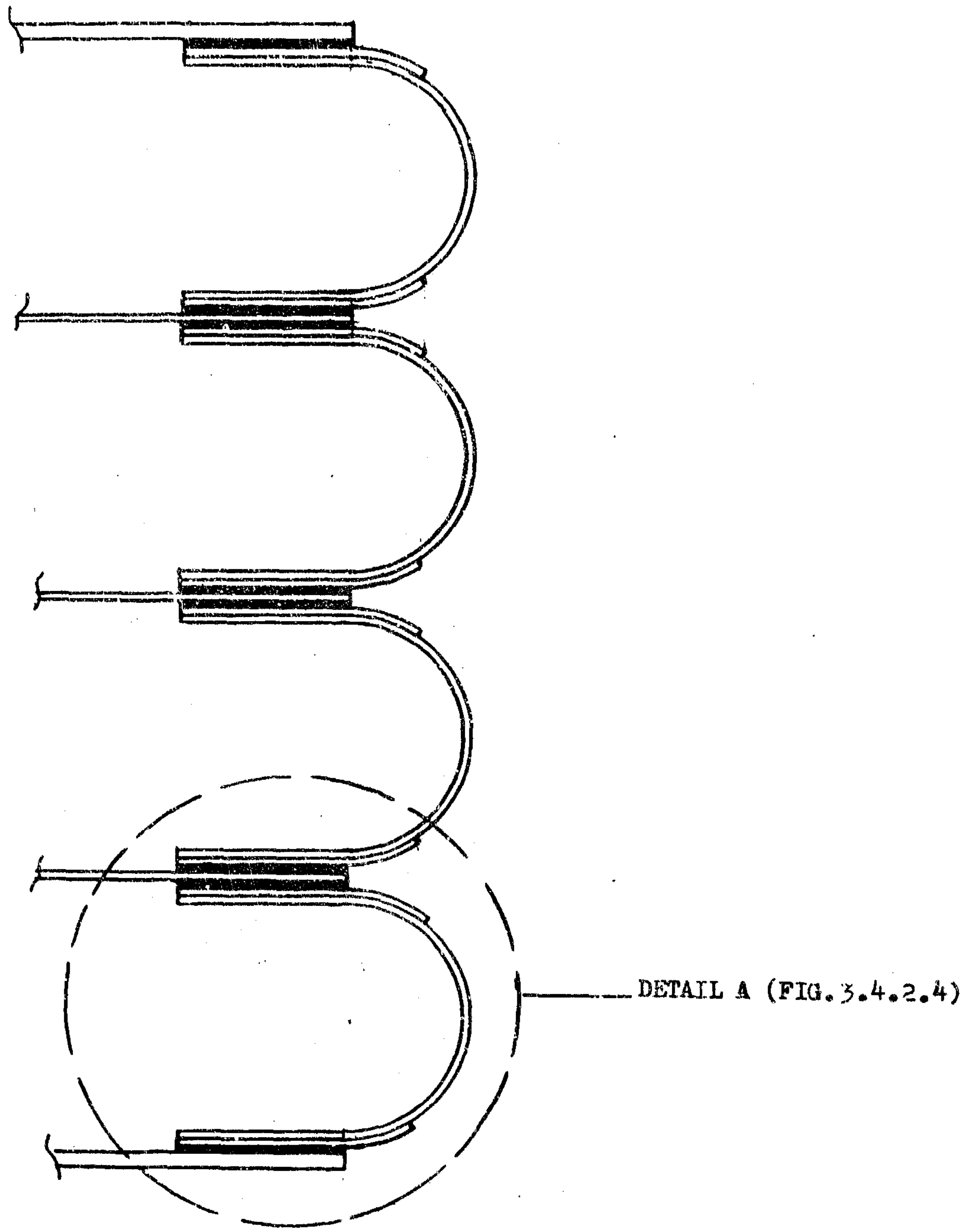
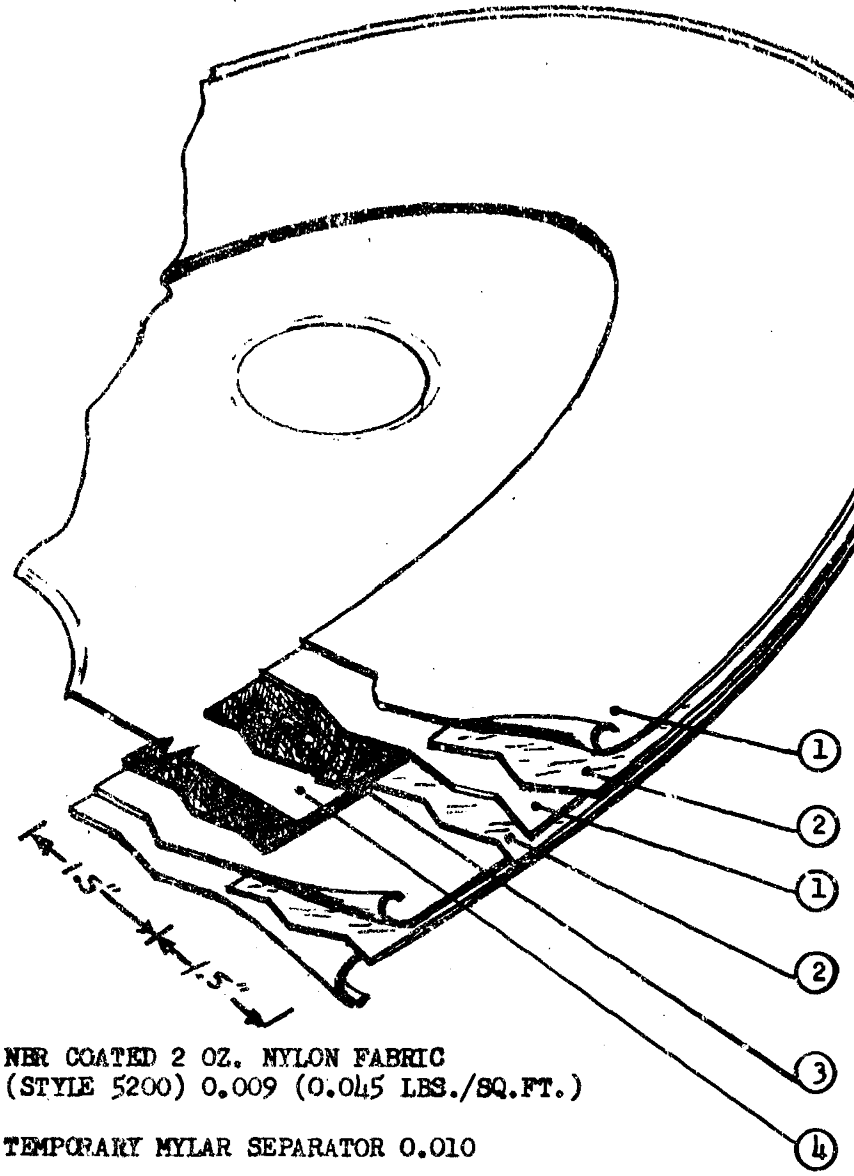


FIG.3.4.2.2

MARTIN CO.
4 MODULE FLOTATION UNIT



① NBR COATED 2 OZ. NYLON FABRIC (STYLE 5200) 0.009 (0.045 LBS./SQ.FT.)

② TEMPORARY MYLAR SEPARATOR 0.010

③ NBR ELASTOMER 3010, 0.010

④ ALUMINUM FORMER, 0.020

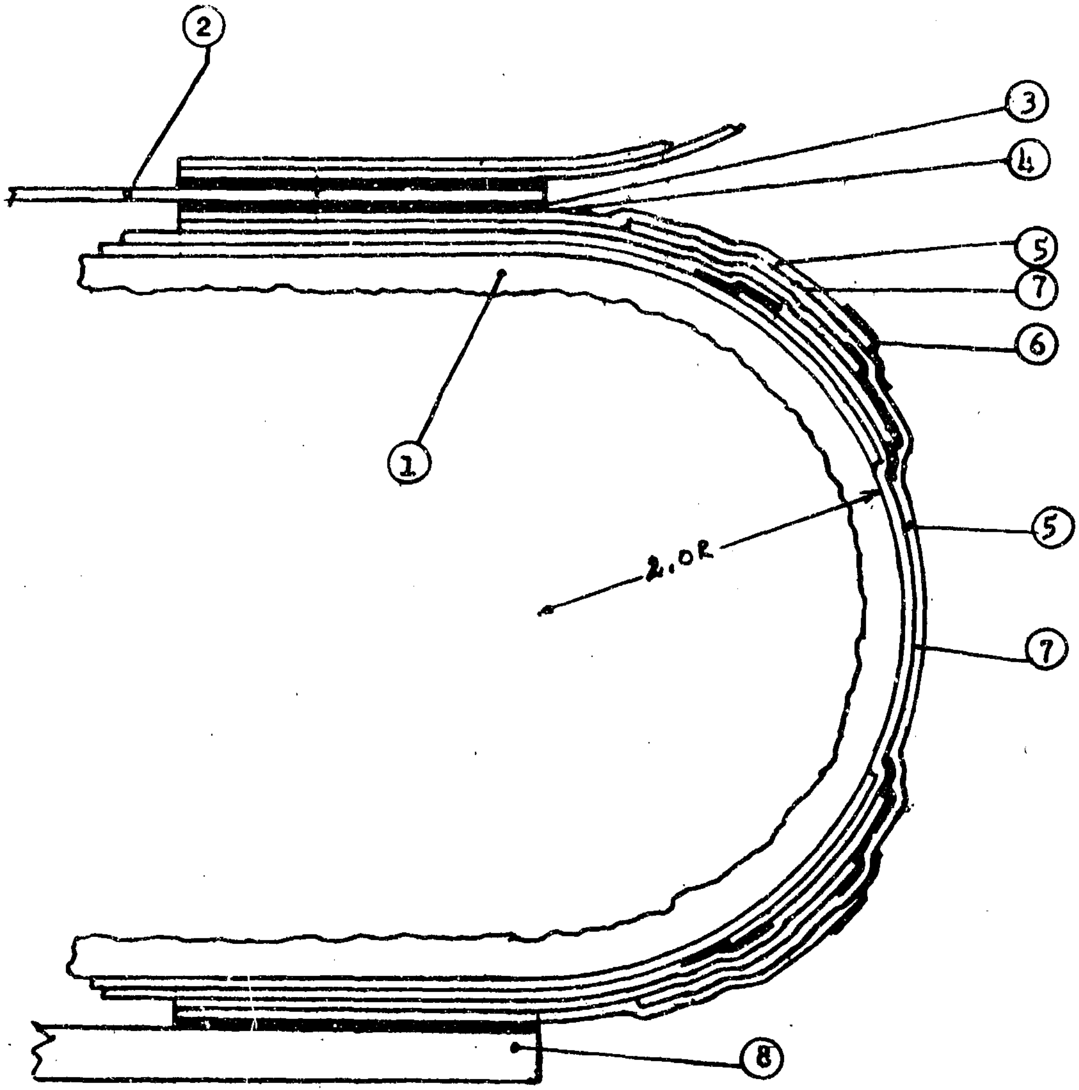
FIGURE 3.4.2.3 MARTIN CO. 4-MODULE FLOTATION UNIT--ALUMINUM FORMER ASSEMBLY

(1) SOAK OUT MANDREL

(2) ALUMINUM FORMER 0.020

(3) NBR ELASTOMER 3010, 0.010

(4) ADHESIVE 3242



(5) NBR COATED 2OZ. NYLON FABRIC (STYLE 5200) 0.009 (0.045 LBS./SQ. FT.)

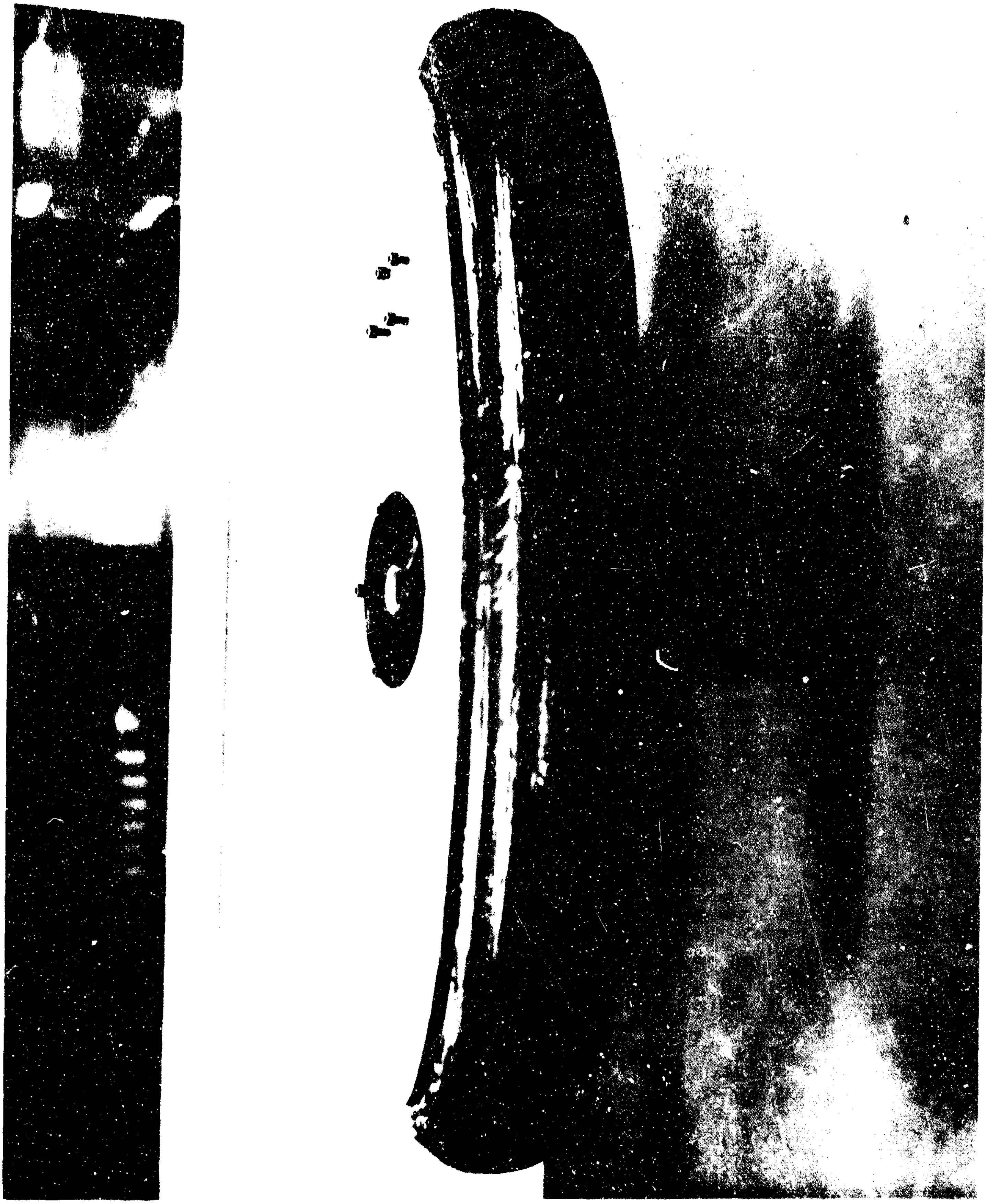
(6) NBR GUM TAPE 5137, 0.010

(7) ADHESIVE 3200

(8) ALUMINUM TERMINAL FORMER 0.25

FIG. 3.4.2.4

MARTIN CO.
4 MODULE FLOTATION UNIT



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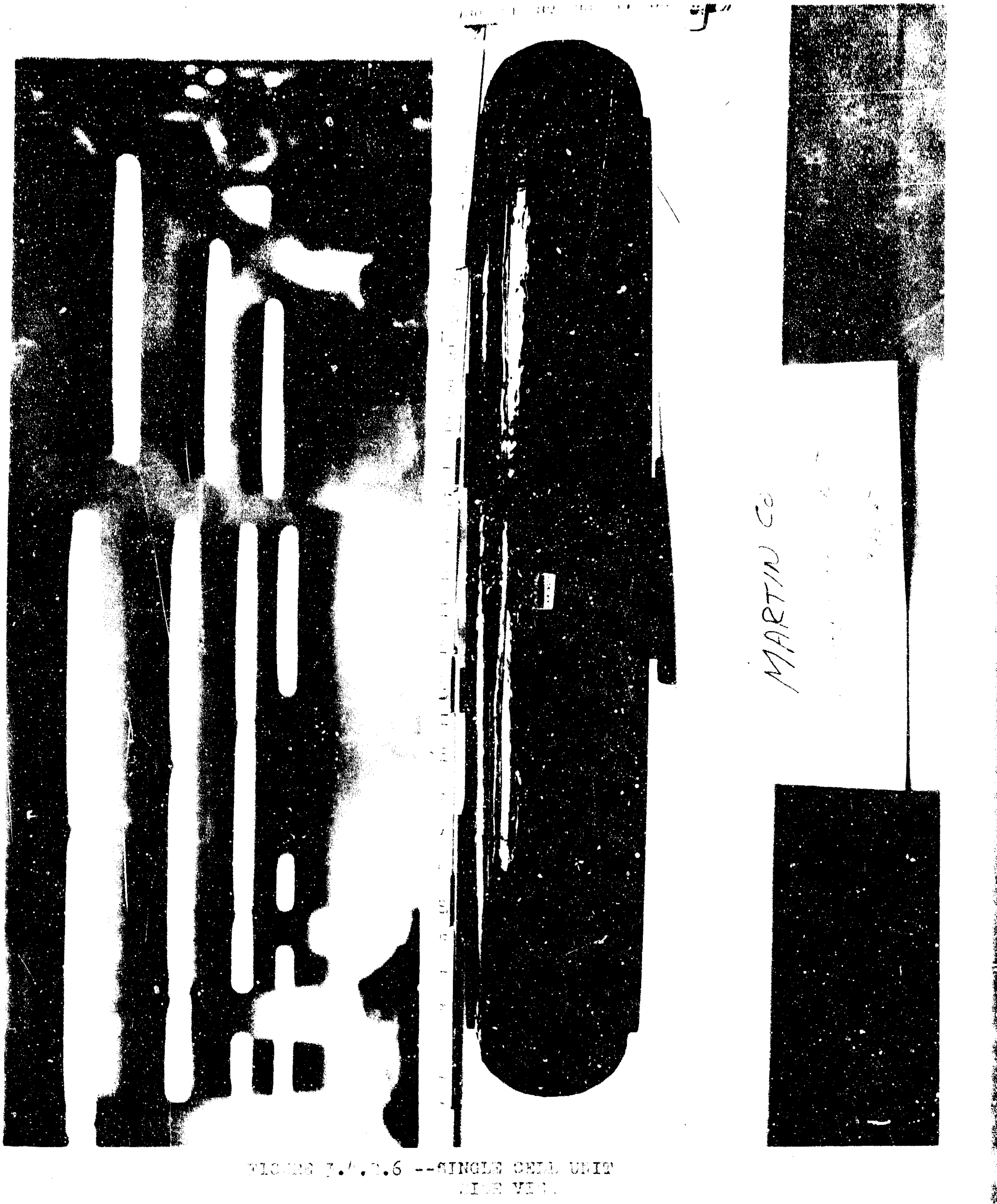
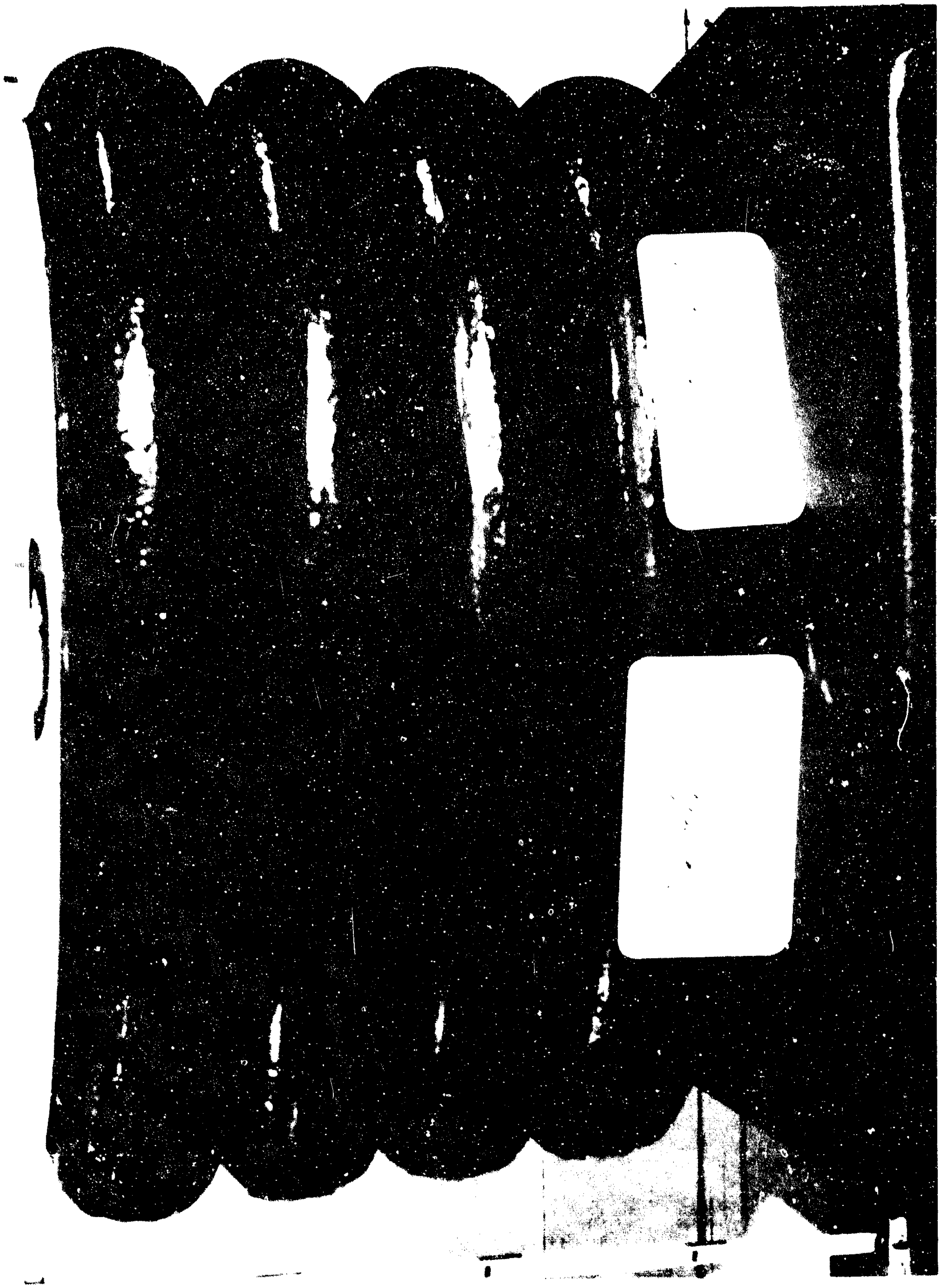
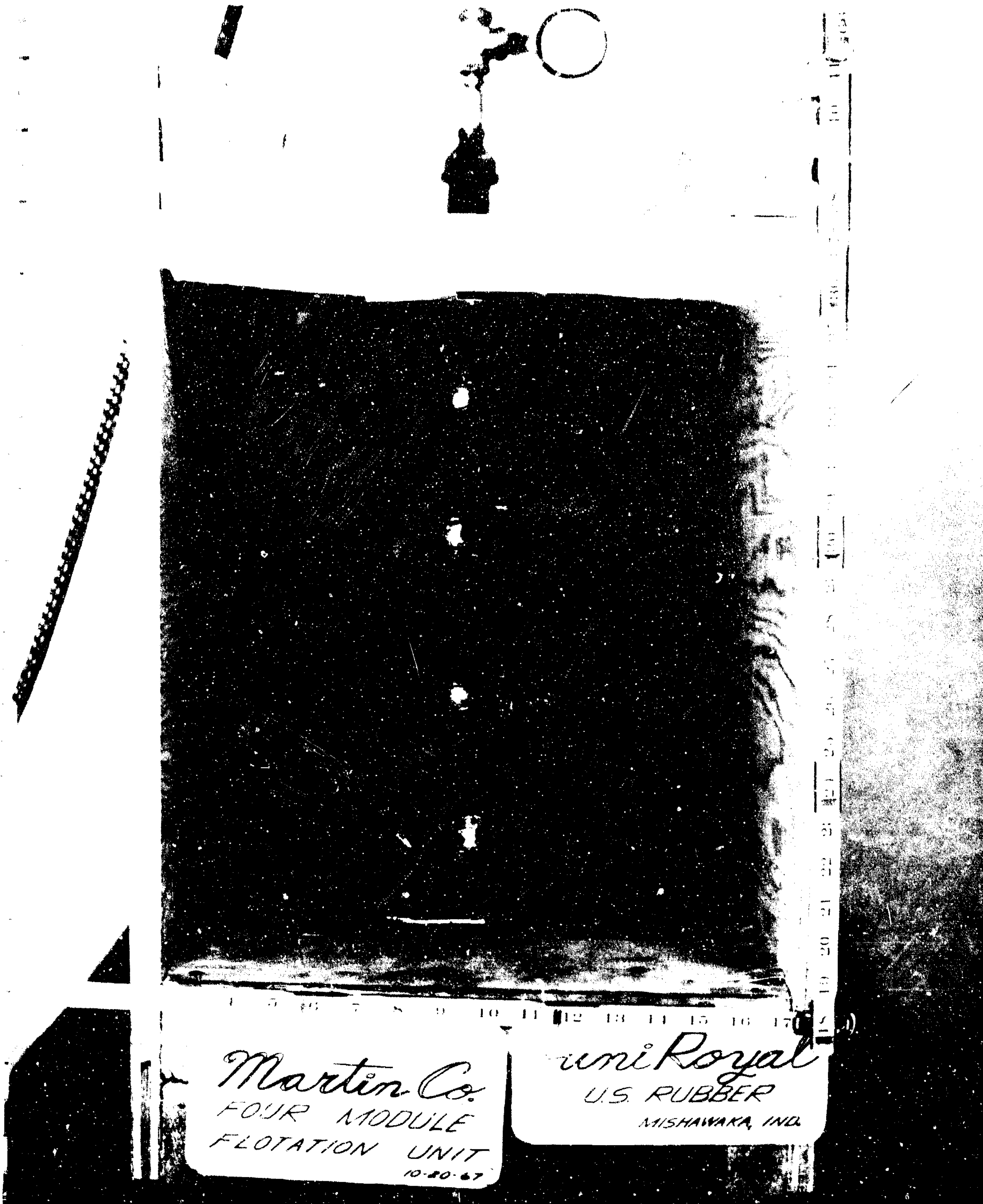
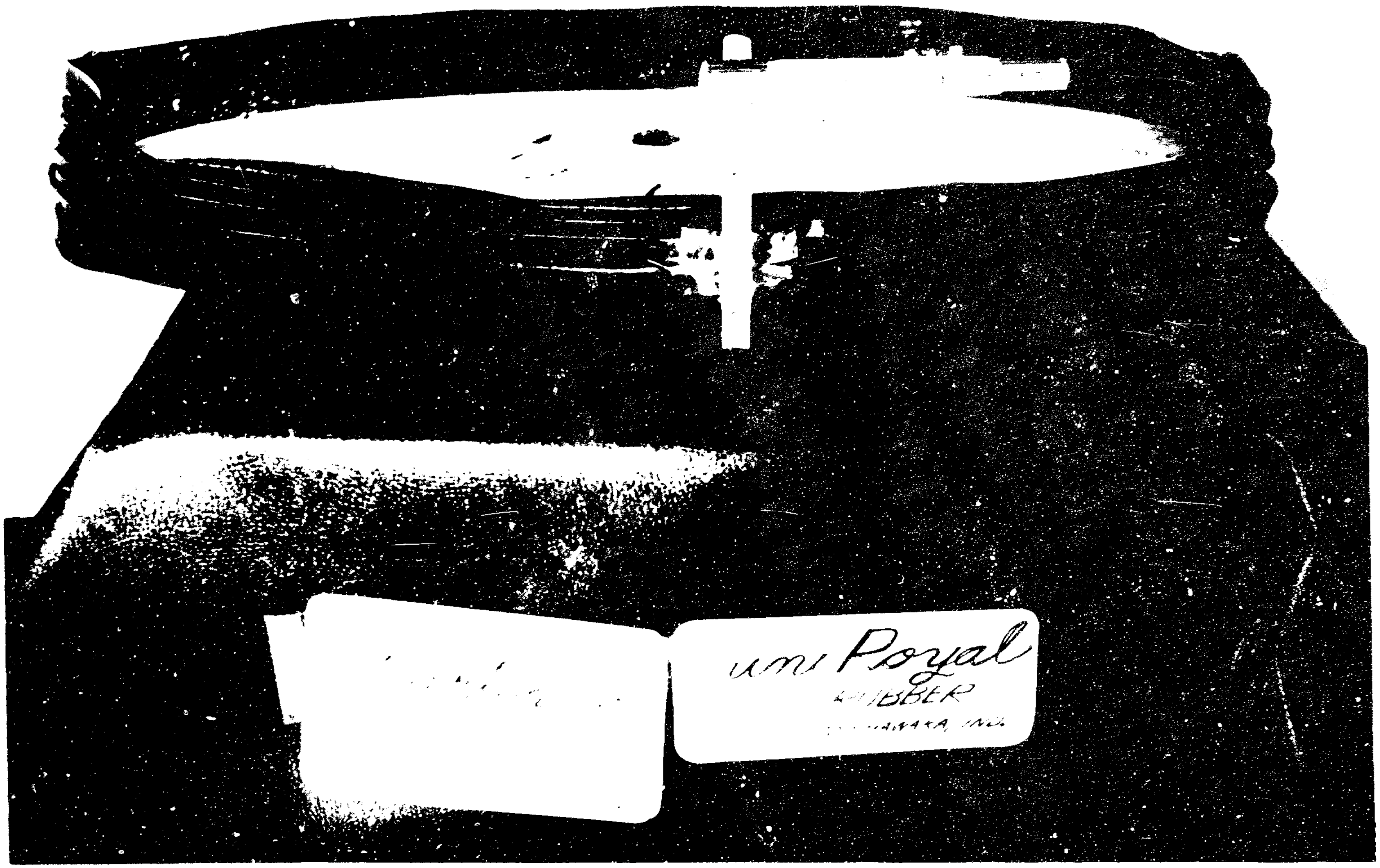


FIGURE 3.4.2.6 --SINGLE CELL UNIT
SIDE VIEW




Martin Co.
FOUR MODULE
FLOTATION UNIT
10-20-67
uniRoyal
U.S. RUBBER
MISHAWAKA, IND.


uni Royal

Paracril OZO coatings. Therefore, a similar system of adhesion for a Neoprene or Paracril OZO coated fabric will be demonstrated should the better aging material be selected for construction of the initial production module assembly.

The actual fabricated and tested four module unit described in this report is somewhat heavier in weight than required due to an oversight during fabrication of not trimming the first flexible wall fabric to the peripheral dimensions shown in Figure 3.4.2.4. Therefore, an excess of 2.0 oz is included in this test unit. This material is unbonded in the area to be cut out and therefore does not alter the seam-bond efficiency.

3.4.3 Float Actuation System

Since a high pressure air system inherently actuates very quickly, an all-pneumatic system was selected to meet the design goal of extending the float system in 30 seconds. Having decided to use a gas system, the problem of providing the stored energy must be considered. Two approaches seemed feasible to obtain the compressed gas. First, atmospheric air could be compressed by an engine driven compressor and then dried and stored in an air tank. This approach would be heavy and result in a complex system to insure dry air. As a result, it was felt that a precharged gas bottle system would be simpler and more reliable. To circumvent the moisture problem (nozzle exit temp. =-15°F), a nitrogen gas system was selected. A schematic diagram of this system is shown in Figure 3.4.3-1. If air were substituted for the nitrogen gas, a drier would have to be incorporated into the system.

A brief study was made to ascertain the storage volume requirements as a function of storage pressure. The basic float system requires 75 cu ft at 5 psig for an extension cycle. This is equivalent to 101 SCF. Figure 3.4.3-2 shows the relationship between storage pressure and volume along with a storage vessel diameter. Three system pressures were evaluated assuming a 3/16" diameter tube and a choked flow orifice. The assumption of choked flow enables one to neglect any back pressure effects. Each system was sized to meet the volume and time requirements for single and double cycle operation. The pertinent results of this study are summarized in Table 3.4.3-1.

Table 3.4.3-1

Storage Press	Cycles	Bottle Dia.	Time (Prel)	No. Bottles
1000 psi	Single	1.46 ft.	39 sec.	2
	Double	1.80 ft.	15, 63 sec.	1
2000 psi	Single	1.15 ft.	22 sec.	2
	Double	1.45 ft.	8, 39 sec.	1
3000 psi	Single	1.04 ft.	17 sec.	2
	Double	1.28 ft.	6, 18 sec.	1

The above numbers are based on no leakage and no contingency. Considering the time element and volume requirements, the 1000 psig system was dropped from consideration. As expected, the 3000 psig system is the smallest and fastest acting. However, according to the data in Reference 14, the 3000 psig bottles have limited availability. Therefore, the 2000 psig system was selected from the inflation system.

As previously mentioned, 101 SCF are required per inflation cycle. At 70°F, the nitrogen gas weighs .00735 lb/ft^3 or a total of 7.42 lbs per cycle. In reality, the system will have inherent leakage. To compensate for this a contingency of 30% by weight was designed into the system. The required weight then is 9.5 lbs of nitrogen per inflation. A single and dual cycle nitrogen system was designed. Typical results of pressure and weight available as a function of time are shown in Figure 3.4.3-3 for the single cycle and Figure 3.4.3-4 for the dual cycle. For the proposed single cycle system, the time required for full inflation is about 21 seconds. The double cycle system will inflate the floats fully in 10 seconds for the first cycle and then requires 32 seconds for the second inflation. Both systems meet the required design goal of 30 seconds.

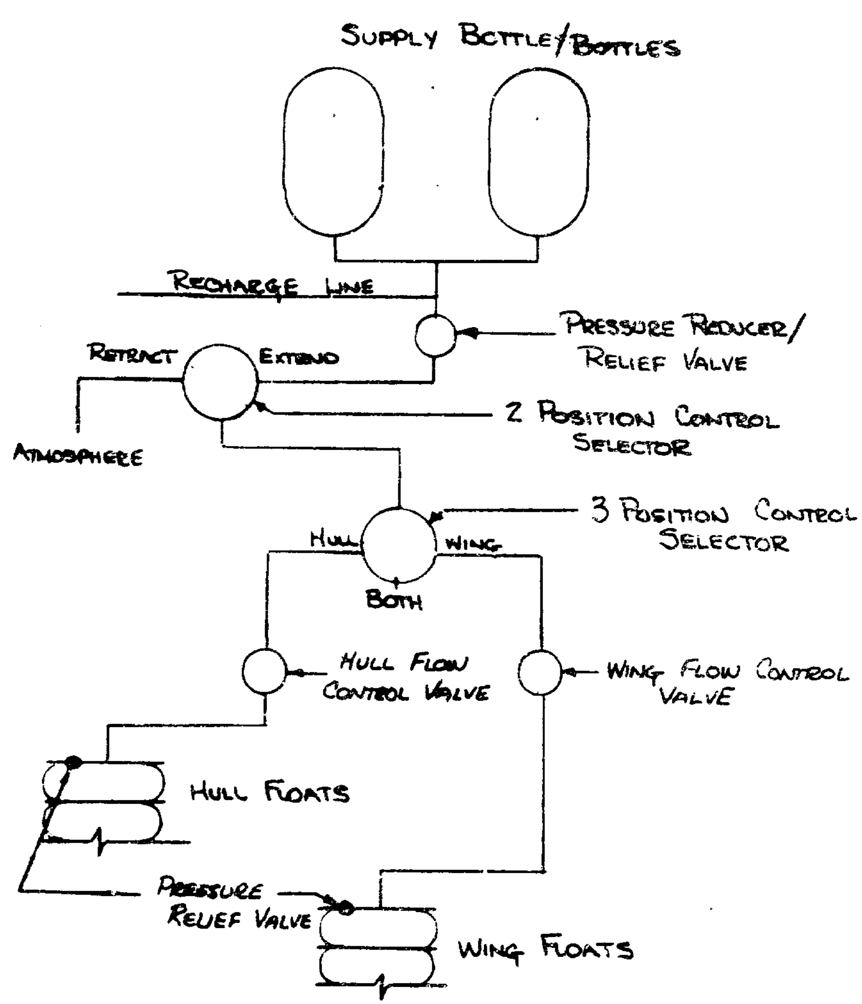
The recommended pneumatic system for the HRV aircraft will be a 2000-psig single nitrogen storage bottle. This will represent the most reliable and lightest weight system. The selection of a 2000-psig system was predicated on having maximum recharge availability. To insure that the floats will reach a final retracted position, a mechanical retraction force was provided. The simplest way to accomplish this was through the use of negator springs (constant force) mounted internally. A value of 80 lbs per float was selected. This then allows approximately 40 lbs to act as a locking type load for getting the floats to their final retracted position.

The system operational sequence is as follows (Reference Figure 3.4.3-1). After the aircraft has been landed and brought to rest on the water surface, the hull floats are rotated into the extension position. The two-position-selector valve is placed on the extend position. Next step is to place the three-position-selector valve on the wing position. With the wing flow control level, the wing floats are allowed to extend into a position which equals the hull float position.

After the wing floats have been positioned, the three-position-selector lever is placed on the both position. Now with one hand on the wing flow control valve and the other on the hull flow control valve, the floats are extended. Any unusual motion of the aircraft will be sensed by the operator and corrected through the appropriate flow control. A full down indicator will then be actuated completing the extension.

For retracting the floats, the two-position-selector lever is placed on the retract position. This will allow the system to vent to the atmosphere. The three-position-selector level will still be on the both position. Now both flow control valves will be opened and the pressure will vent. In addition to the pressure venting action, two forty lb negator

VERTICAL FLOAT - INFLATION SYSTEM DIAGRAM



INFLATION SYSTEM
CHARACTERISTICS

SUPPLY PRESSURE – 2000 psig

GAS ~ NITROGEN

TWO INFLATION CYCLE CAPACITY

FLOAT VOLUME REQ'D = 75.1 CU. FT. @ 5psig
(101 STD. CU. FT.)

FIGURE 3.4.3 - 1 INFLATION SYSTEM

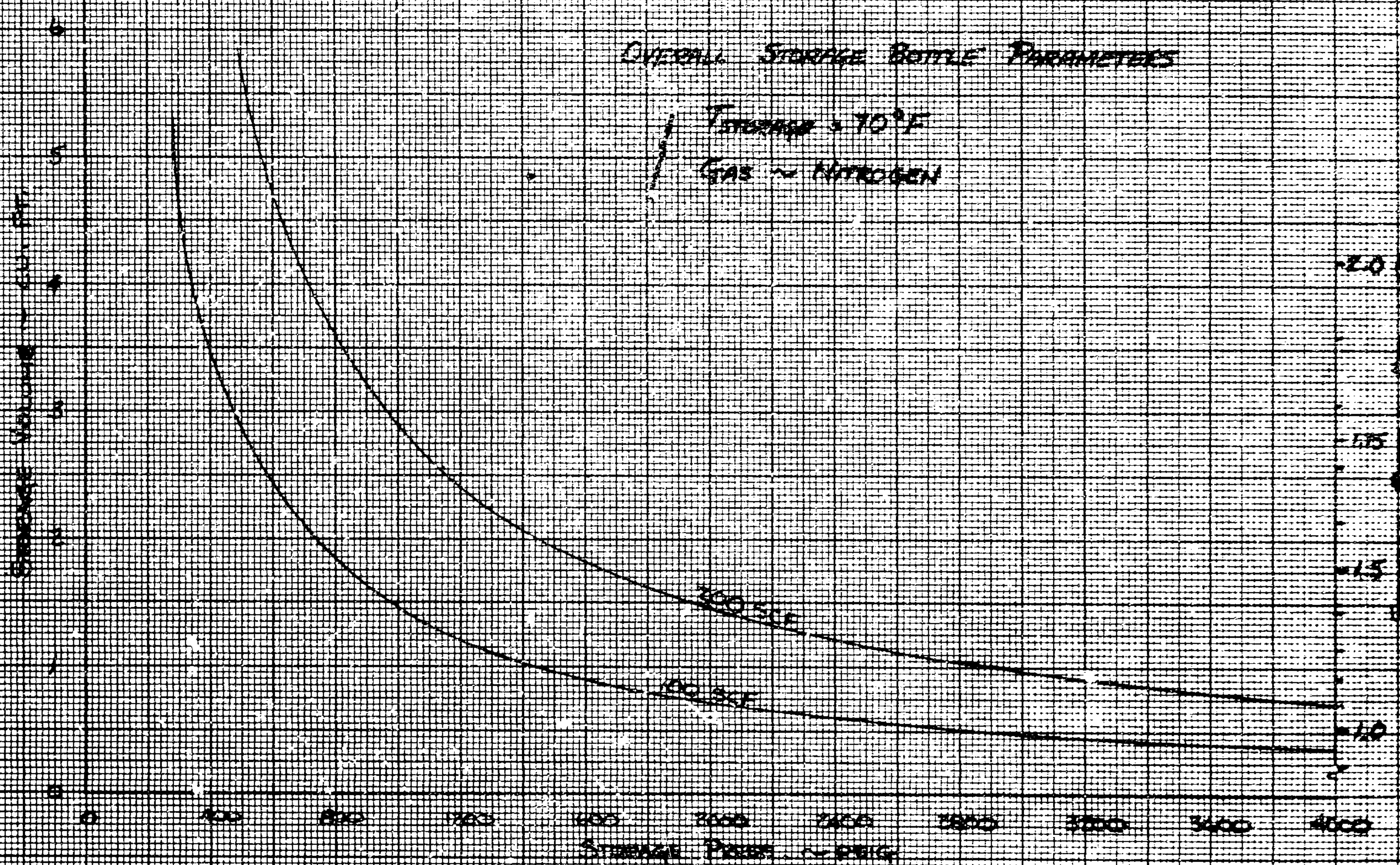


Figure 3.4.3-1. - Storage Bottle Pressure-Volume Parameters.

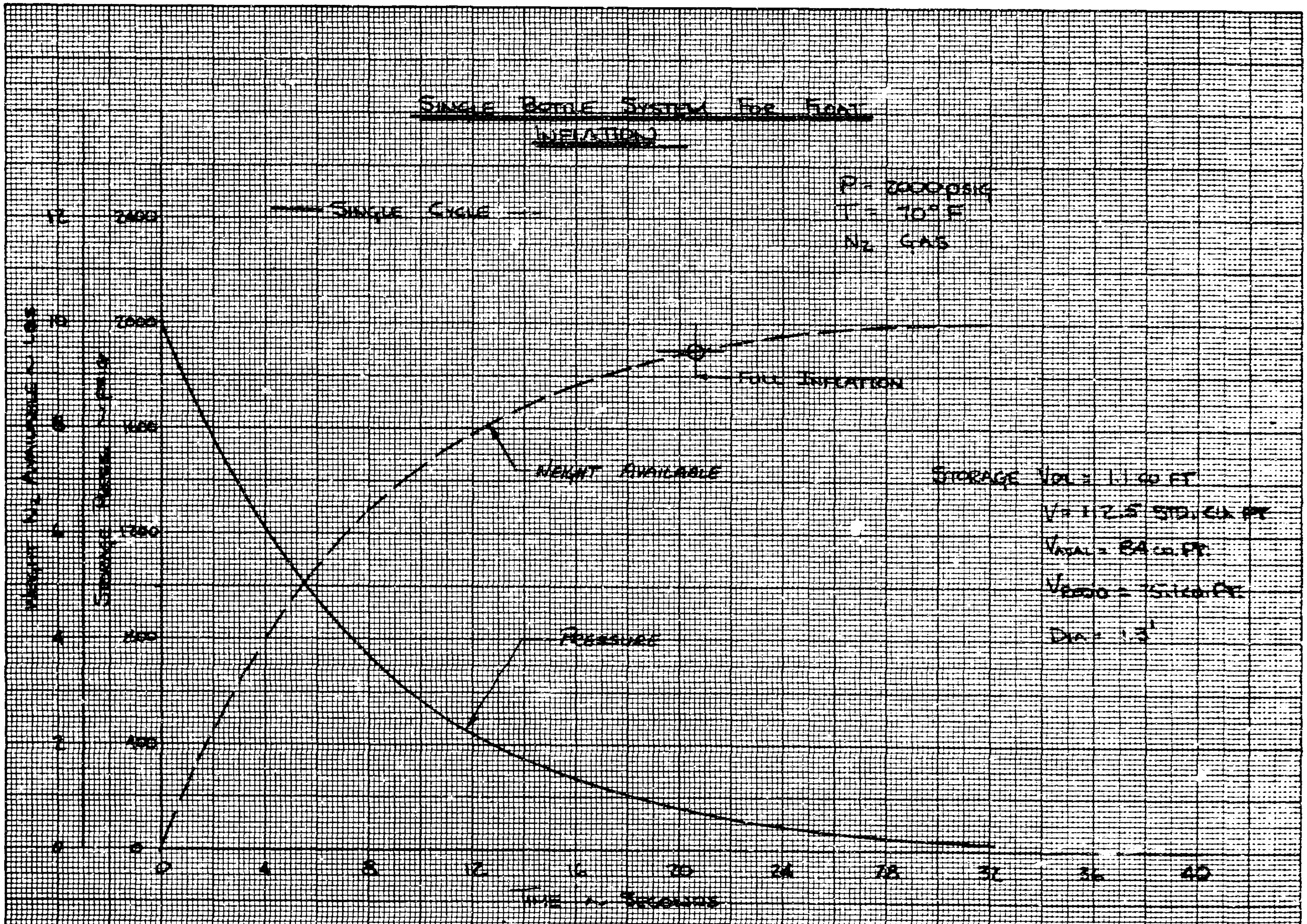


FIGURE 3.4.3-3 TYPICAL SINGLE CYCLE SYSTEM

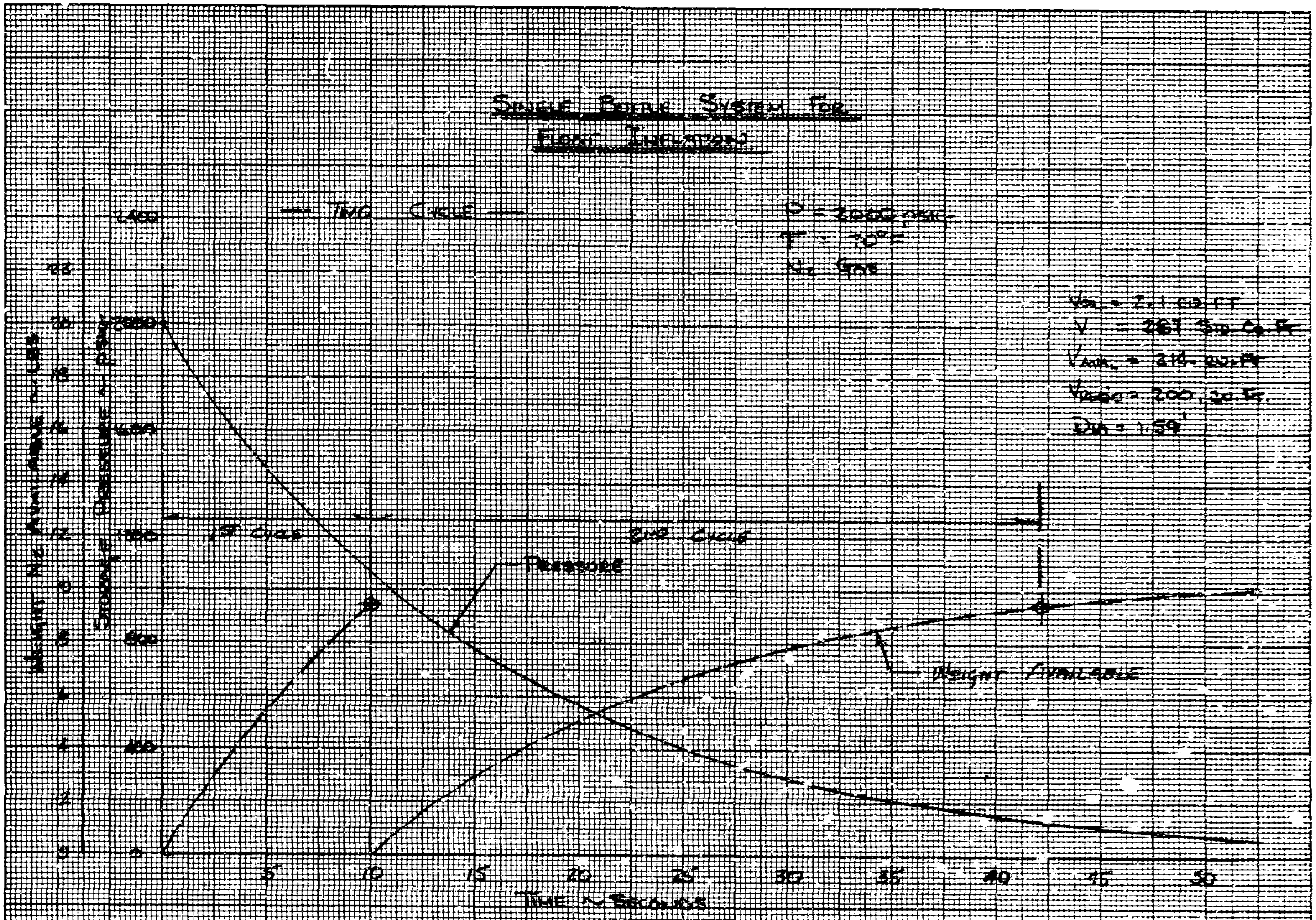


FIGURE 3.4.5-4 TYPICAL DUAL CYCLE SYSTEM

springs per float help to bring the system back into the final retracted position. Once the floats have been retracted, the hull floats are rotated into the flight position and the aircraft is prepared to take off.

3.4.4 Weight Summary

An evaluation of the Vertical Float System was made to assess the weight. The present design was based on keeping the manufacturing costs to a minimum. This approach will always yield a heavier system. In order to indicate what might be accomplished by a weight reduction effort, a column of potential weights is shown. Table 3.4.4-1 summarizes the present and potential weight breakdown.

Table 3.4.4-1

WEIGHT SUMMARY

Item	Demonstration System	Potential System
Float Weight		
Support Tube	22.1	17.6
Intermediate Ribs	15.3	13.4
End Ribs	1.5	1.4
Uniroyal Bag	3.2	3.2
Damping Plate	4.7	3.8
Hull Fittings	1.8	1.7
Total per Float	48.6	41.1
Total for 6 Floats	291.6	246.6
System Weight		
Storage Bottle*	60.0	40.0
Tubing	5.4	4.5
Pneumatic Components	5.0	4.0
Sub-system Total	362.0	295.1
HRV Modifications		
Structural / Mechanical ** }	35.0	31.0
TOTAL	397.0	326.1

*Reference Figure 3.4.4-1
**Estimated

The total system weight is within the established design goal.

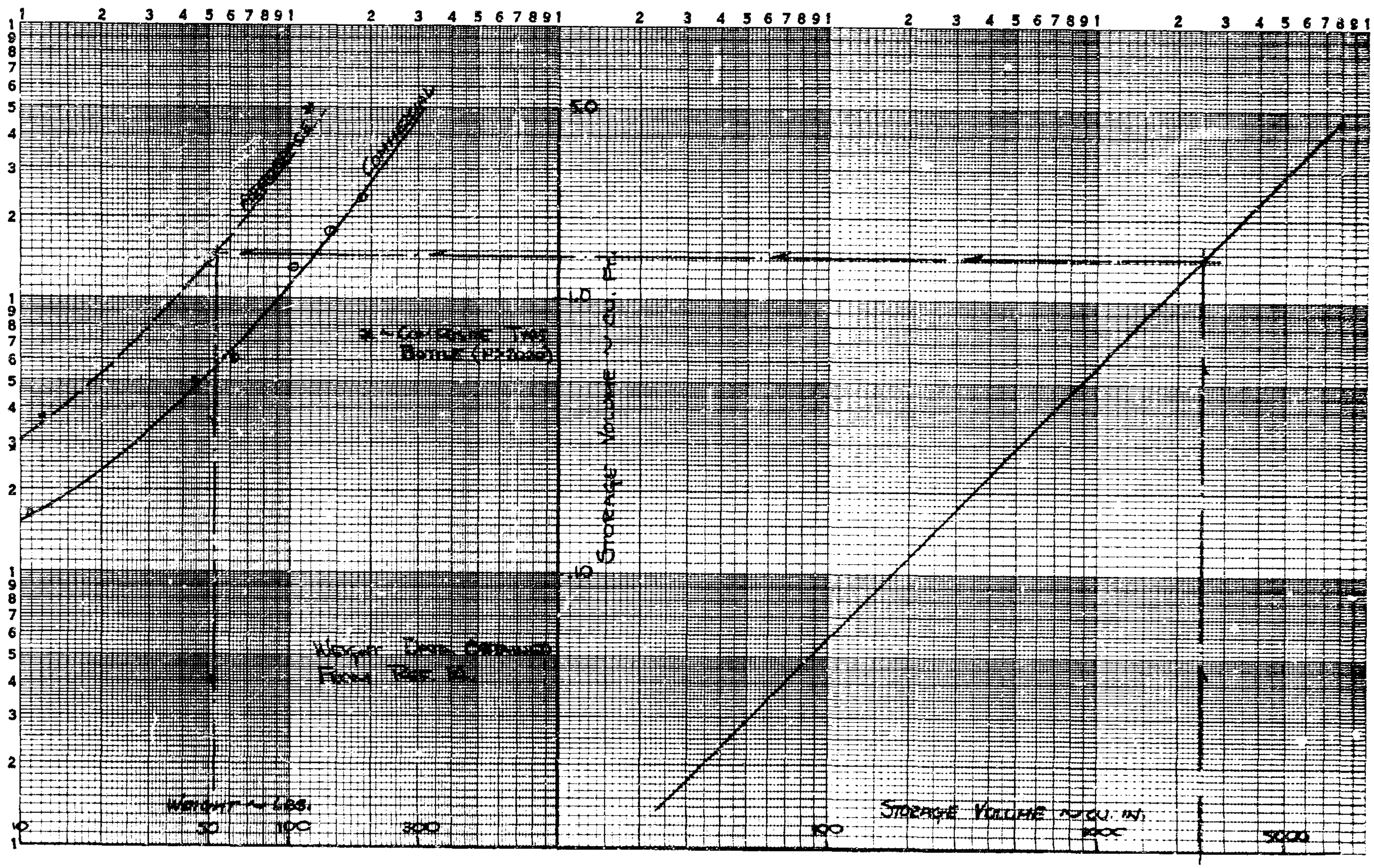


Figure 3.4.4-1 ~ Storage Bottle Volume-Weight Relationship

4.0 AIRCRAFT MODIFICATION

The incorporation of the Vertical Float System on the HRV aircraft will create condition for which the airplane was not designed. Since the float stations were located at anticipated structural hard points, the overall structural strength requirements should not change significantly. In this phase of the contract, only the general arrangement was established as depicted on Drawing No. 818171. However, the Phase II or early Phase III engineering effort must evaluate in detail the influence of the Vertical Float System loads. The following paragraphs discuss some of the major studies which must be completed on the HRV airframe.

The most obvious study for structural integrity will be a comprehensive evaluation of overall float-airplane loading conditions. Having derived these, comparisons can be made of Vertical Float System loading conditions with existing strength.

Local structural problems will involve specialized design condition or self-equilibrating load systems due to mechanical motion requirements. For the Vertical Float System, the first category of local problems effects both wing and hull floats, while only the hull floats will require rotation into the retracted position. Both wind and hull floats will experience take-off and landing water loads. These loads must have a primary path through the float structure. To accomplish this, a positive locking device will be used. The wind float in its retracted position must provide structure to accommodate the wing float loads. Although engineering intuition indicates that airloads will not provide any critical structural requirements, the airload conditions should be checked. The mechanical type design considerations will be most severe for the hull floats (Reference Section A-A, Drawing No. 818171). Here, the problems of actuator support structure, lock support structure, etc., must be designed along with checking existing structure for any induced mechanical loads.

Compromise situations may occur where the HRV airplane could be restricted to a given operational envelope in order to minimize structural weight penalties.

The installation of the inflation system should pose no severe problems. In most cases, the access availability may create some special considerations. With all the incorporated weight items, it may be necessary to evaluate the HRV weight and balance effects on flight performance and stability.

5.0 PROPOSED PROGRAM FOLLOW-ON

The proposed program will be accomplished as follows. Phase II will consist of initial engineering of the complete Inflatable Vertical Float System for incorporation on the HRV aircraft and a development test of one float and associated systems. A detailed outline of the Phase II effort may be found in Reference 20. Phase III will consist of final engineering of the complete float system, the incorporation of the complete system on the HRV aircraft, and a demonstration test of the aircraft and float system.

Phase II will carry the Inflatable Vertical Float concept through the layout design stage for each float and all associated systems for their deployment and stowage. Layout designs will be accomplished to define the necessary modifications to the HRV airplane for incorporation of the float system. One full scale float assembly will be built and tested to hydrodynamic loads while immersed in water at various angles and depths. The test will demonstrate not only that the buoyancy forces can be adequately supported without excessive distortion or deflection of the flotation bag of the supporting telescoping column but also that the float system can be repeatedly extended and retracted without bending failure or excessive wear. The test specimen will be affixed to a structural steel fixture situated on an existing steel platform on the Martin Marietta Corporation dock or on an open-well barge. See Figure 5.1.

Phase III will comprise final engineering incorporating changes resulting from the development testing of Phase II, the modification of the HRV airplane, and demonstration testing of the airplane and float systems. The modifications of the HRV airplane will consist of installing the six float assemblies, two on the bow, two on the hull afterbody, and one on each wing, replacing the existing wing tip floats. The installations will include systems for deployment and stowage of the floats and also associated cockpit control. All instrumentation and equipment required to document the demonstration testing will be installed. The demonstration will consist of actual water landings, float deployment, retraction and stowage, and airplane takeoffs. The tests will be done in smooth and rough water to illustrate complete operational feasibility of the Vertical Float concept.

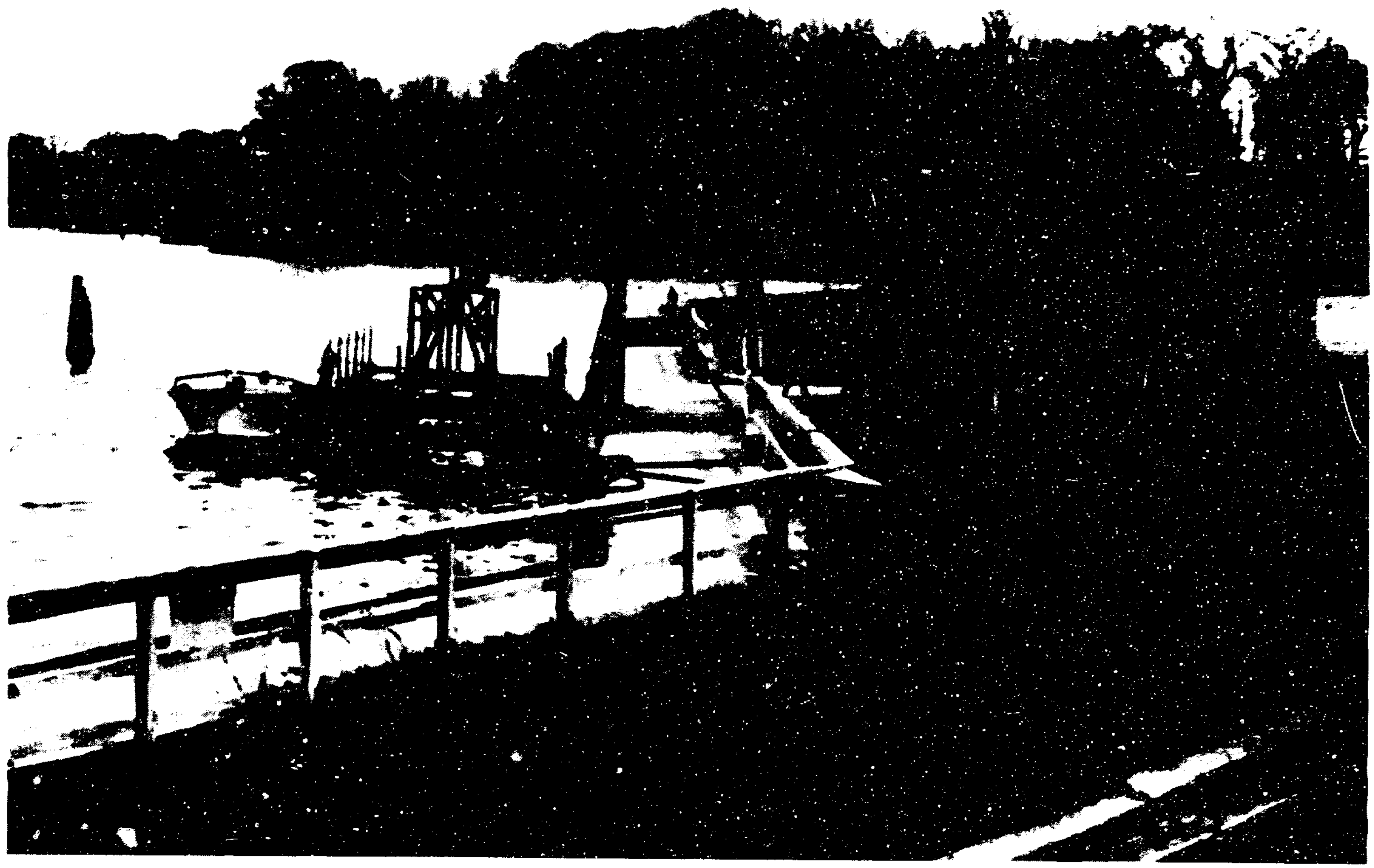


Fig. 5.1. The Proposed Test Site

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APPENDIX A

SUMMARY OF TEST RESULTS ON FLUID COLUMN TYPE CONCEPT

SUMMARY

A pin ended fluid column assembly was tested under externally applied axial loads.

The assembly included a reinforced rubber hose approximately four inches inside diameter clamped to aluminum alloy end fittings, and filled with water at various pressure levels. The overall length was about four feet eight inches.

The externally applied column loads were equal to about eighty-five percent of the axial internal pressure loads for incremental internal pressures up to four hundred pounds per square inch. The maximum external load was about forty-three hundred pounds without column failure.

At internal pressures above four hundred pounds per square inch, slippage of the end fittings introduced considerable beam column bending so that the tests had to be discontinued. The failing strength of the column could not be determined.

The effective modulus of elasticity of the reinforced rubber hose was estimated from strain gage readings recorded during the tests. The estimated value in the axial direction is one hundred fifty thousand pounds per inch. This value is somewhat in question because of slippage of the end fittings during the tests.

The weight of the reinforced hose was less than five pounds and quite flexible when unpressurized.

The installation is shown in photographs in the back of this report.

INTRODUCTION

The following test was conducted using a rubber fluid column assembly 56-3/16 inches long. The assembly included a reinforced rubber hose, 49.5 inches long with an O.D. of 4.22 inches, manufactured by the U. S. Rubber Company. The reinforcement was manifest by proprietary longitudinal and circular winding patterns of plastic strands. The resulting hose was flexible in a relaxed condition. The hose ends were secured to aluminum hardware with steel clamps. The hardware was designed to form a fluid seal and machined to receive fluid injection tubes and ball ends which were mated with socket blocks when the assembly was mounted for test. An exploded view of the end hardware is shown in photographs B-84317 and B-84318.

Three sets of strain gages (one transverse and one longitudinal per set) were attached to the hose 120° apart on a circumferential line midway from each end.

Additional details pertinent to the assembly are shown below:

1. Total weight of rubber fluid column with end hardware attached but without ball ends - 23.5 lbs.
2. Weight of ball ends - 2.398 lbs. each (sockets not included).
3. Weight of one end fitting without ball end - 9.35 lbs.
4. Weight of hose alone - 4.8 lbs.
5. Length of hose - 49-1/2 inches.
6. Circumference of hose - 13-1/4 inches.
7. O.D. of hose - 4.22 inches.
8. I.D. of hose - 3.97 inches.
9. Hose wall thickness - 1/8 inches.
10. Overall length of column with ball ends attached prior to installation in the test machine - 56-3/16 inches.

PROCEDURE

The hose assembly was filled with water, then installed vertically in a 400,000 pound Baldwin-Lima-Hamilton Universal Testing Machine. A socket block was provided to receive each ball end. The installation is shown in photographs B-84265 and B-84266. A hydraulic pump and pressure gage were connected to the assembly to provide the various levels of internal pressure desired. The vertical load was applied by use of the test machine. Appropriate instrumentation provided both transverse and longitudinal strain readings.

Strain readings were taken at internal pressure increments of 100 psi up to 400 psi with vertical load increments of 200 pounds up to 200 psi and 250 pounds beyond 200 psi. The load limits were 1000, 2200, 3000 and 4300 pounds for internal pressure lvels of 100, 200, 300 and 400 psi respectively.

RESULTS

The longitudinal strain readings obtained at various internal pressure levels and corresponding loads are tabulated in Table No. I. Figures 1 through 4 show load-strain curves developed from the tabulated data. Figures 5 and 6 show curves displaying the relationship of internal pressure and applied load respective to circumferential strain.

The initial misalignment was estimated to be about one-quarter of an inch. However, the pressure cycling caused slippage of the end fittings that resulted in continuous increase in misalignment. The bending induced by the increased misalignment limited the load carrying capacity of the assembly to 4300 pounds. Had there been no increase in misalignment it is believed the failing load would have been much higher.

DISCUSSION

Since some column bending was detected in the longitudinal direction, it was necessary to separate the bending and axial effects which, due to the placement of the strain gages were recorded as one by the strain indicator. This would then permit a determination of the modulus of elasticity presumably exhibited by the column material. Toward this end, the strain indicator readings in the longitudinal direction were evaluated with sights focused on an average elongation or compression which the hose might be considered to experience without bending (i.e., transforming the bending load into an equivalent axial load), between successive loads.

Example: Internal pressure at 100 psi

Load (lbs)	Strain (μ in/in x 10)		
	Gage 3L	Gage 1L	Gage 2L
200	- 8	+30	+47
400	-16.5	+18	+37.5

In view of the bending and the signs of the strain readings, it is evident that the neutral axis passes through a plane such that gages 1L and 2L are on one side and gage 3L is on the opposite side. Assuming that the strain is linear around the outer edges of the hose, there should exist a strain midway between gages 1L and 2L of +38.5 μ in/in x 10 in the case of the 200 pound load and one of +27.8 μ in/in x 10 in the case of the 400 pound load.

In view of the strain gage locations (120° apart on the same cross section), these two strain readings are expected to be on the same diameter with the respective strain readings of gage 3L, which makes the situation analogous to the problem of a hollow cylinder being subjected to a pure bending load at its midlength, except for the fact that the stresses on the nose are not equal and opposite.

Consider a diameter through gage 3L for a 200 pound load. The strain readings on either end are -8×10^{-5} in/in and $+38.5\times10^{-5}$ in/in. If we assume that a strain of $+30.5\times10^{-5}$ in/in reflects a difference in stretching of one side over the other, the stresses due to bending can be considered equal and opposite and the strain is an indication of an elongation or compression in the axial direction. For a 400 pound load, the difference in stretch of one side over the other is 27.8×10^{-5} in/in $-$ 16.5×10^{-5} in/in or $+11.3\times10^{-5}$ in/in.

Consider the two successive axial strain indications for the two successive loads. To determine the modulus of elasticity, it is now necessary to convert axial load to stress and plot it as a function of axial strain and develop a curve, the slope of which is the modulus. The modulus E would be,

$$E = \frac{\text{stress}}{\text{strain}} = \frac{\dfrac{400-200}{\text{Area}}}{+30.5-(+11.3)} \times 10^5 = \frac{\dfrac{400-200}{\pi D}}{30.5-11.3} \times 10^5$$

$$= \frac{\dfrac{200}{12.56}}{19.2} \times 10^5 = .83\text{x}10^5 \text{ lbs/in}$$

This is the modulus in the longitudinal direction in lbs/in.

Figure 5 represents the transverse strain reactions at three points on the same cross section, 120° apart, due to pressure variations within the hose. The transverse strain gage is not affected by bending loads. Therefore the strain readings are considered representative of a condition of pure radial pressure. The modulus then can be read directly from the internal pressure vs strain curve.

REFERENCE

ER 14230F, April 29, 1966, An Inflatable Vertical Float System.

Table I

LONGITUDINAL STRAIN READINGS

(100 psi Internal Pressure)

Load (lbs)	Gage Readings (in/in x 10^{-5}) 3L	1L	2L	Diff. in Avg. Rdgs.		Modulus (as measured by $\frac{\Delta \text{ Load} \times 10^5}{\Delta \text{ Avg. Rdgs.}}$)
						$Ex10^5$
200	- 8	+30	+47	+30.5		
					19.2	10.5
400	-16.5	+18	+37.5	+11.3		
					19.8	10.0
600	-24	+ 4	+27	- 8.5		
					5.0	40.0*
800	-32	- 8	+16.5	- 3.5		
					21.0	9.6
1000	-40	-21	+ 6	-24.5		

Avg. 10.03
E = $.795 \times 10^5$ lb/in.

(200 psi Internal Pressure)

Load (lbs)	3L	1L	2L	Diff. in Avg. Rdgs.		Modulus
200	+50	+98	+108	+85.3		
					14.0	14.3
400	+38	+78	+ 98	+71.3		
					12.3	16.3
600	+28	+62	+ 87	+59.0		
					10.7	18.7
800	+19	+49	+ 77	+48.3		
					10/3	19.4
1000	+10	+37	+ 67	+38.0		
					9.3	21.5
1200	+ 3	+26	+ 57	+28.7		
					11.2	17.9
1400	- 5	+15	+ 48	+27.5		
					17.0	11.8
1600	-12	+ 5	+ 40	+10.5		
					8.5	23.5*
1800	-20	- 6	+ 32	+19.0		
					15.0	13.3
2000	-27	-15	+ 25	+ 4.0		
					13.0	15.4
2200	-33	-25	+ 20	- 9.0		

Avg. 16.51
E = 1.32×10^5 lb/in.

*These data not utilized.

Table I (contd)

LONGITUDINAL STRAIN READINGS

(300 psi Internal Pressure)

Load (lbs)	Gage Readings (in/in x 10^{-5}) 3L	1L	2L	Diff. in Avg. Rdgs.		Modulus (as measured by $\frac{\Delta \text{Load} \times 10^5}{\Delta \text{Avg. Rdgs.}}$)
2000	+ 7	+74	+110	+63.7		
					3.8	65.7*
2250	- 5	+52	+ 93	+67.5		
					25.7	9.73
2500	-16	+34	+ 81.5	+41.8		
					22.8	10.95
2750	-27	+20	+ 72	+19		
					22.5	11.10
3000	-38.5	+ 6	+ 64	- 3.5		

Avg. 10.59
E = .845x10^5 lb/in.

(400 psi Internal Pressure)

Load (lbs)	3L	1L	2L	Diff. in Avg. Rdgs.		Modulus
3000	+20	+85	+130	+78.3		
					+15.3	16.3
3250	+ 8	+65	+116	+63.0		
					+ 8.5	29.4*
3500	- 5	+47	+106	+71.5		
					+14.0	17.9
3750	-17	+32	+ 97	+57.5		
					+32.5	7.7*
4000	-30	+21	+ 89	+25.0		
					+19.0	13.2
4250	-42	+14	+ 82	+ 6.0		

Avg. 15.80
E = 1.26x10^5 lb/in.

*These data not utilized.

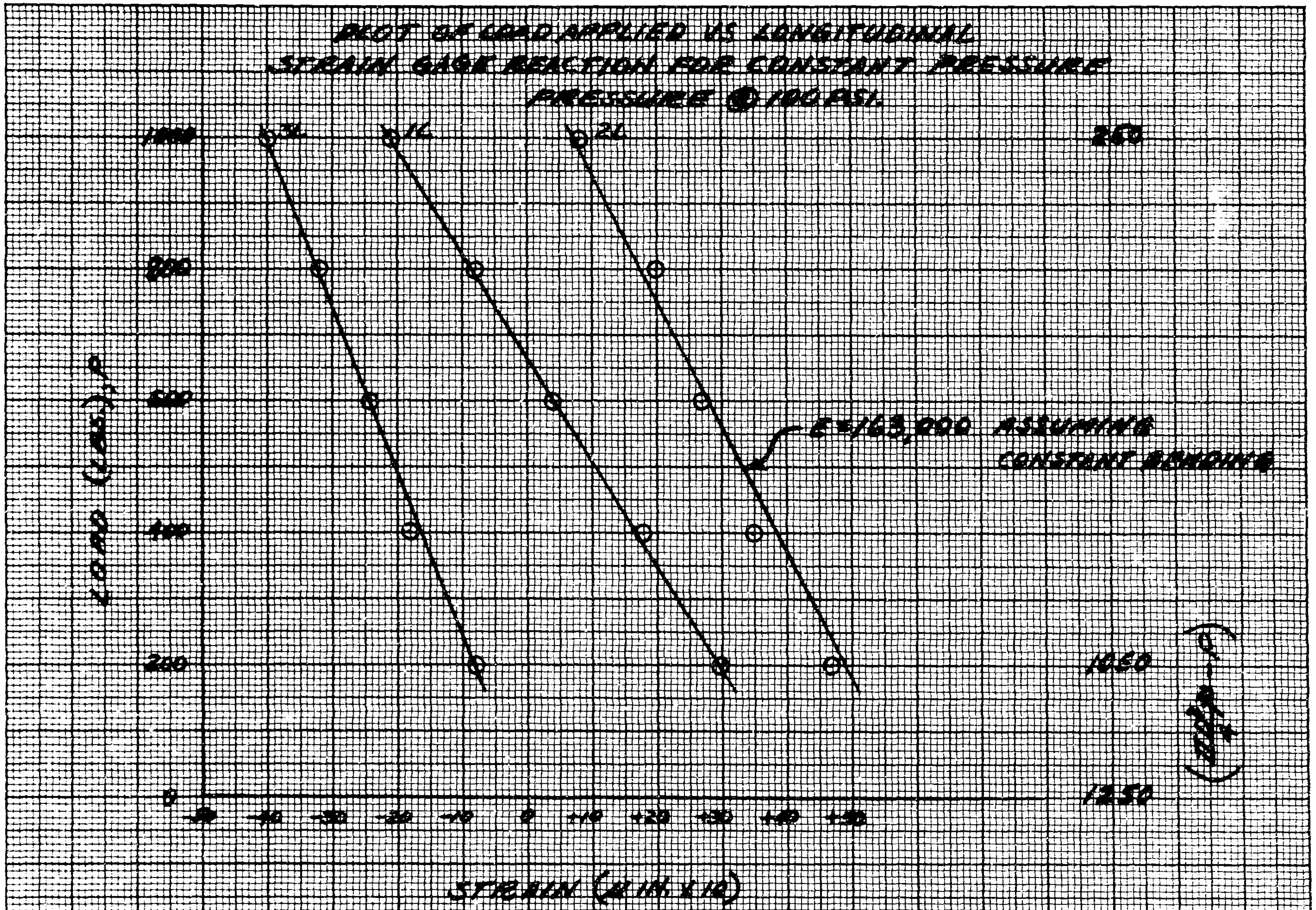

PLOT OF LOAD APPLIED VS LONGITUDINAL
STRAIN GAGE REACTION FOR CONSTANT PRESSURE
PRESSURE @ 100 PSI.
3L
1L
2L
E=163,000 ASSUMING
CONSTANT BENDING
LOAD (LBS.), P
1000
800
600
400
200
0
-50
-40
-30
-20
-10
0
+10
+20
+30
+40
+50
250
1050
1250
STRAIN (μ IN. x 10)

FIGURE NO. 1

FORM # 070286 K+E ALBANENE® TRACING PAPER 74681

PLOT OF LOAD APPLIED VS LONGITUDINAL STRAIN GAGE REACTION FOR CONSTANT PRESSURE OF 300 PSI.

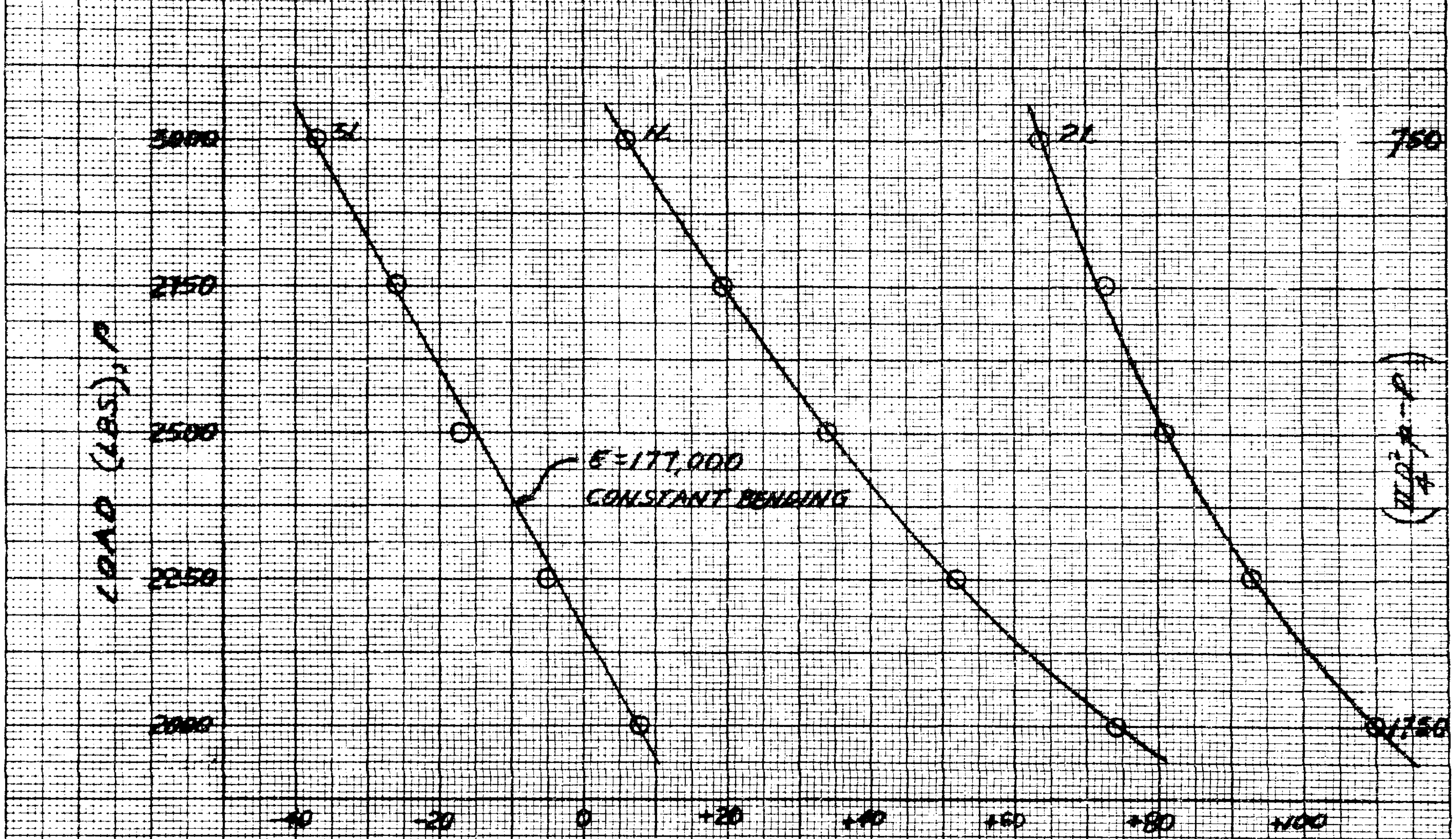
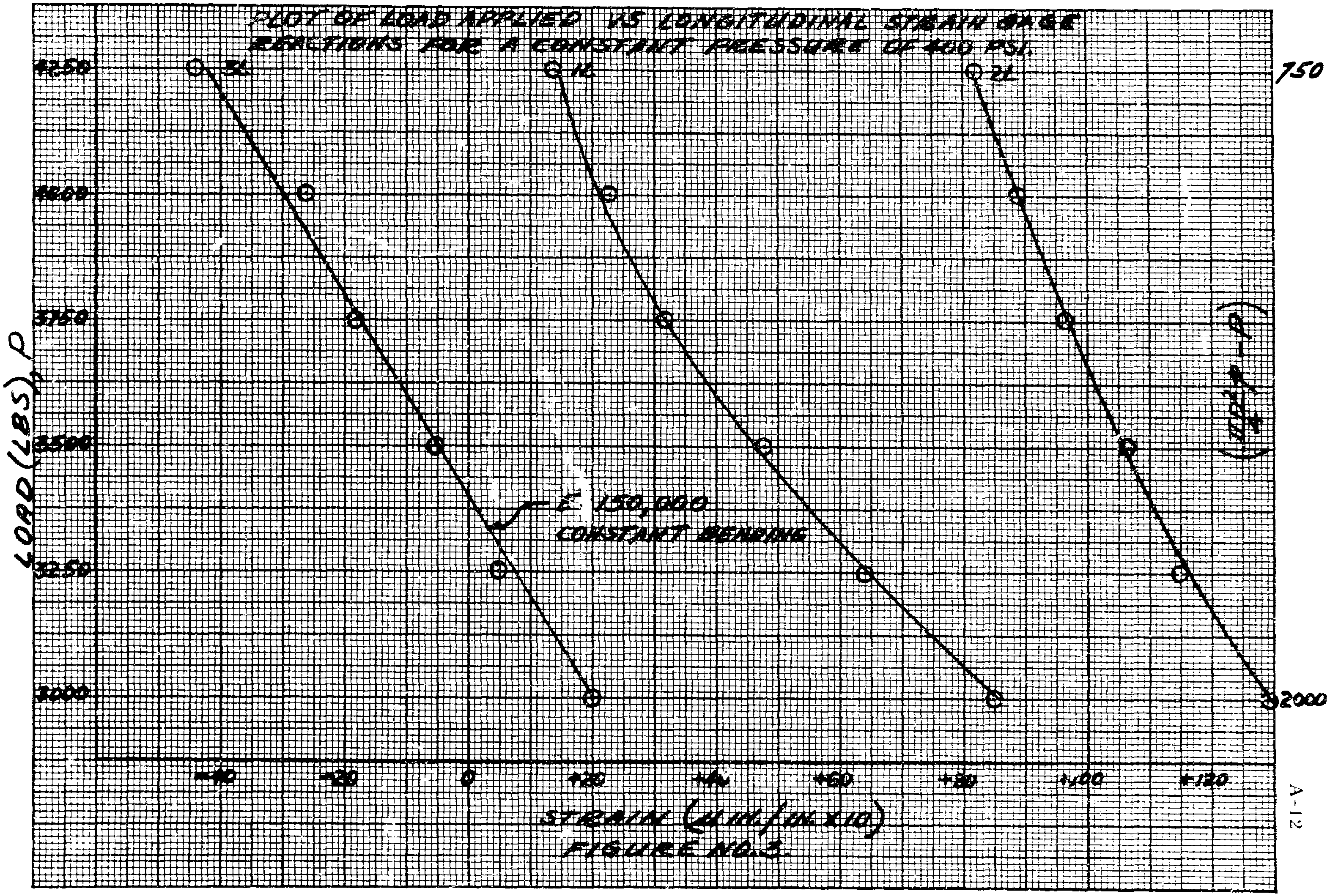
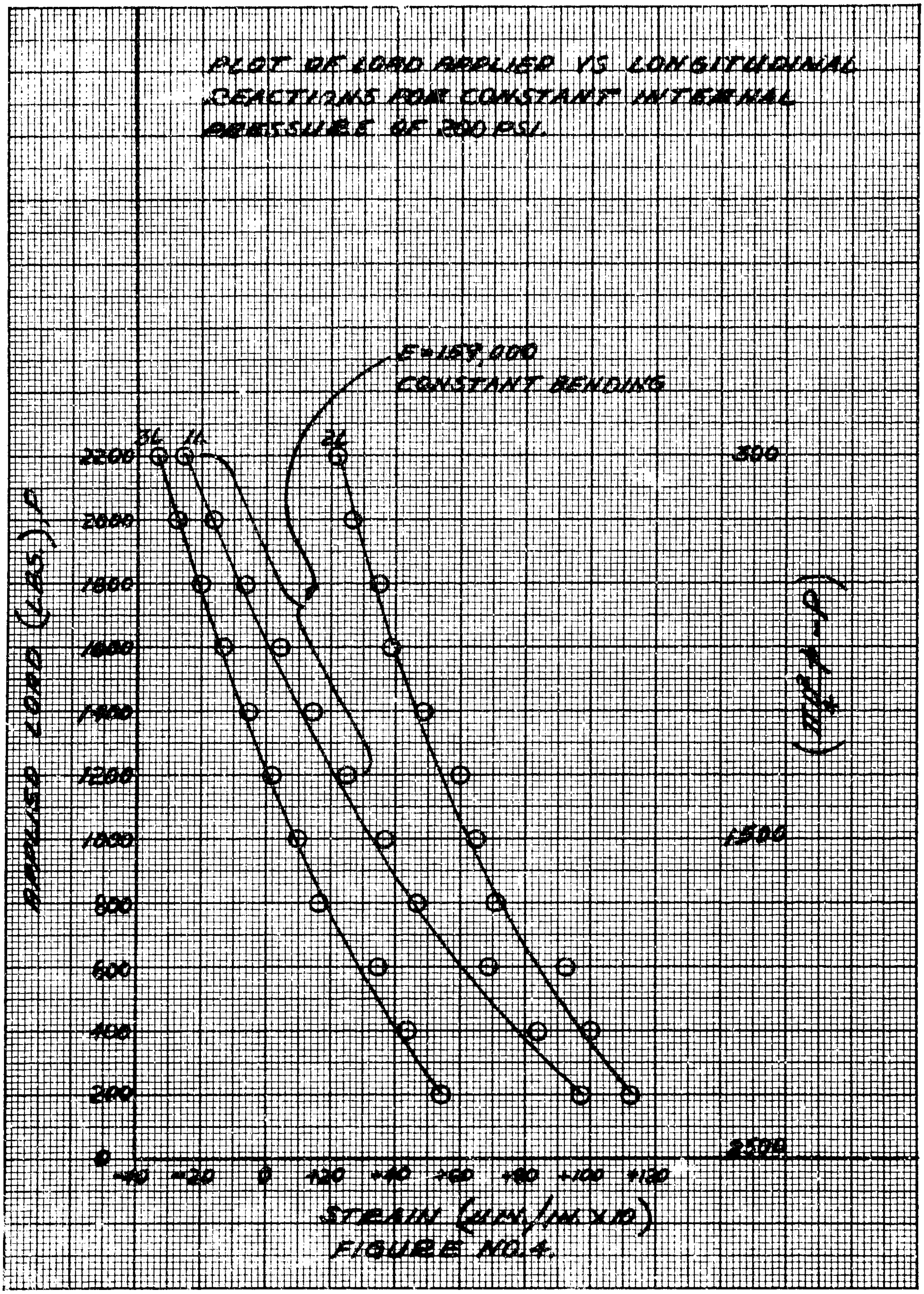
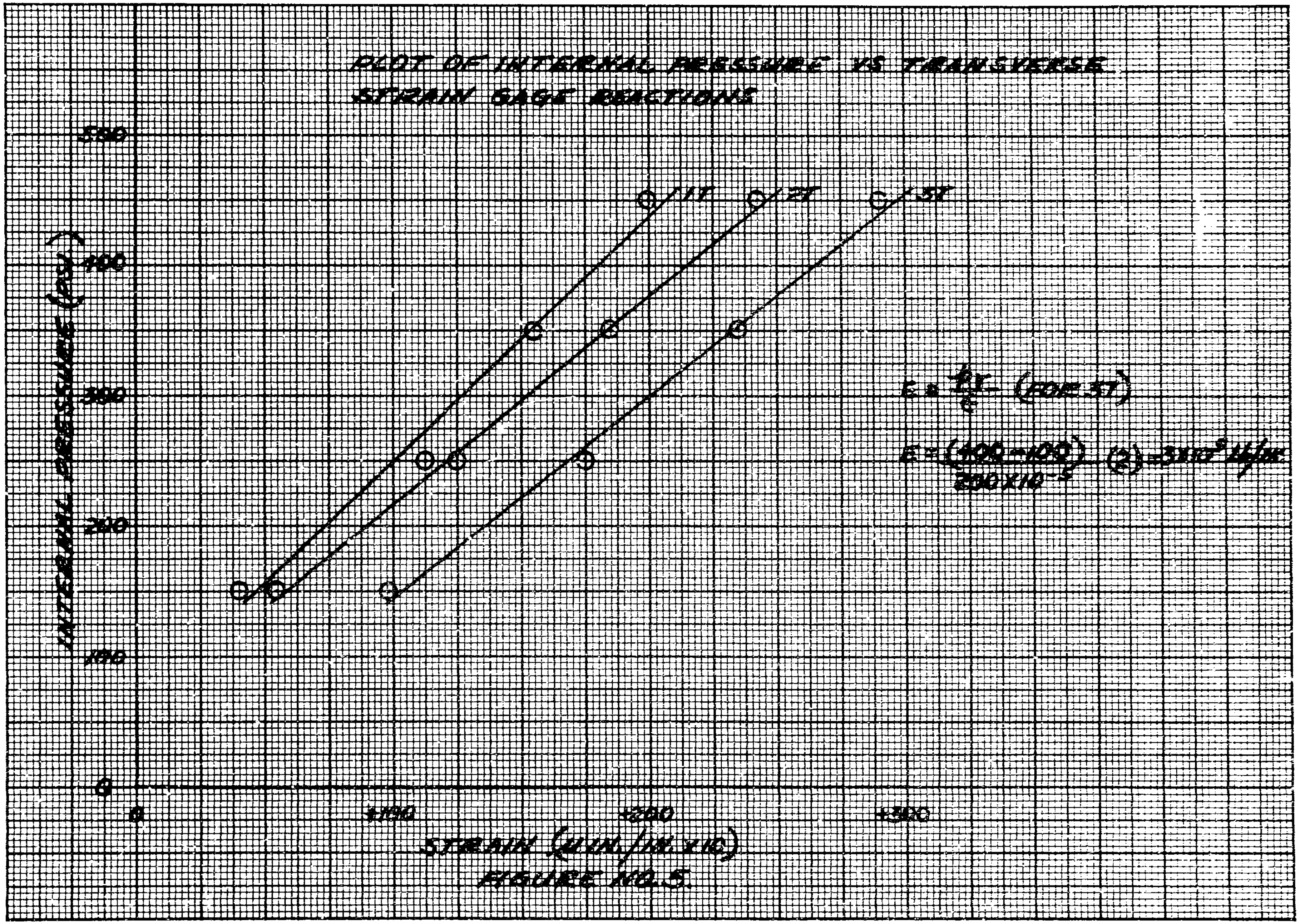


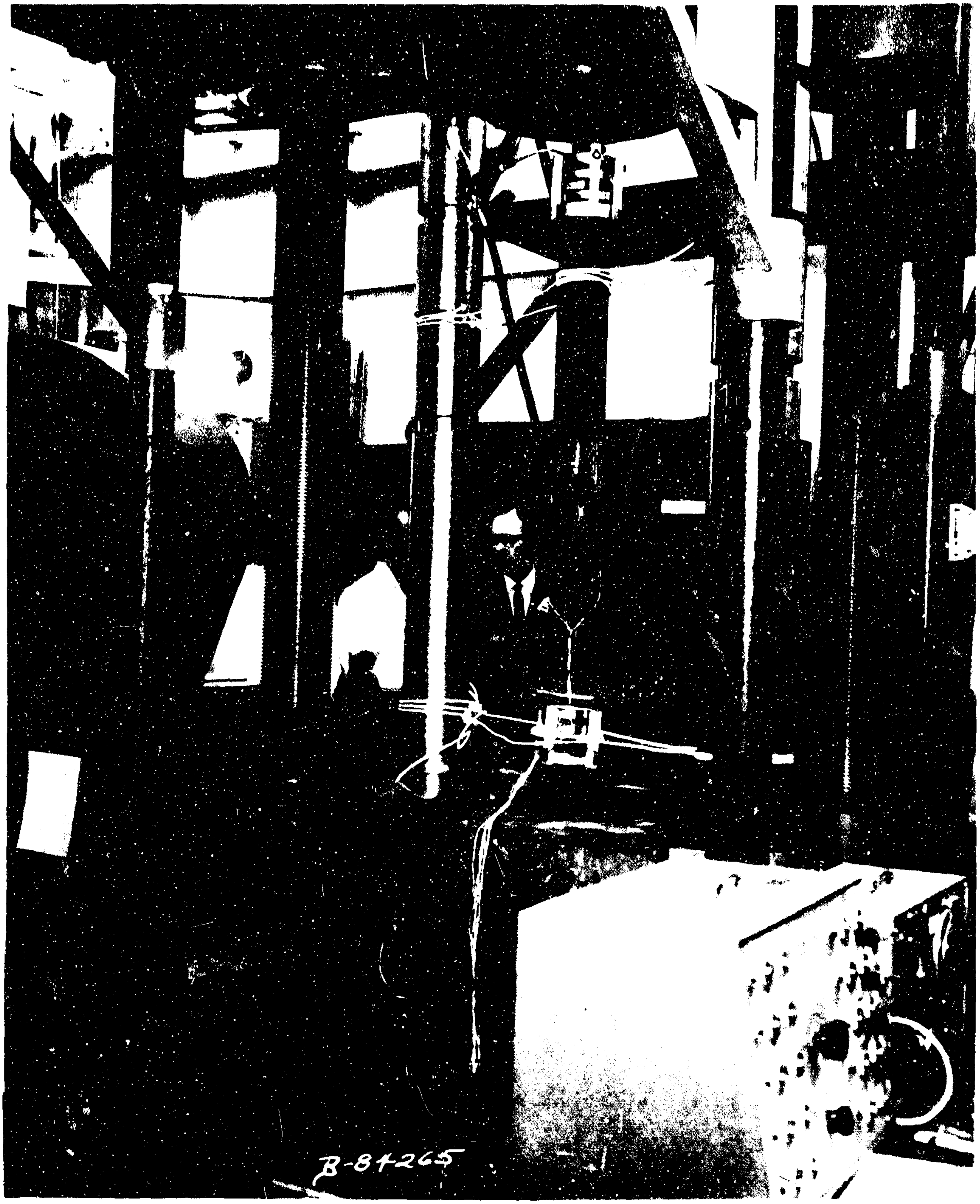
FIGURE NO. 2.

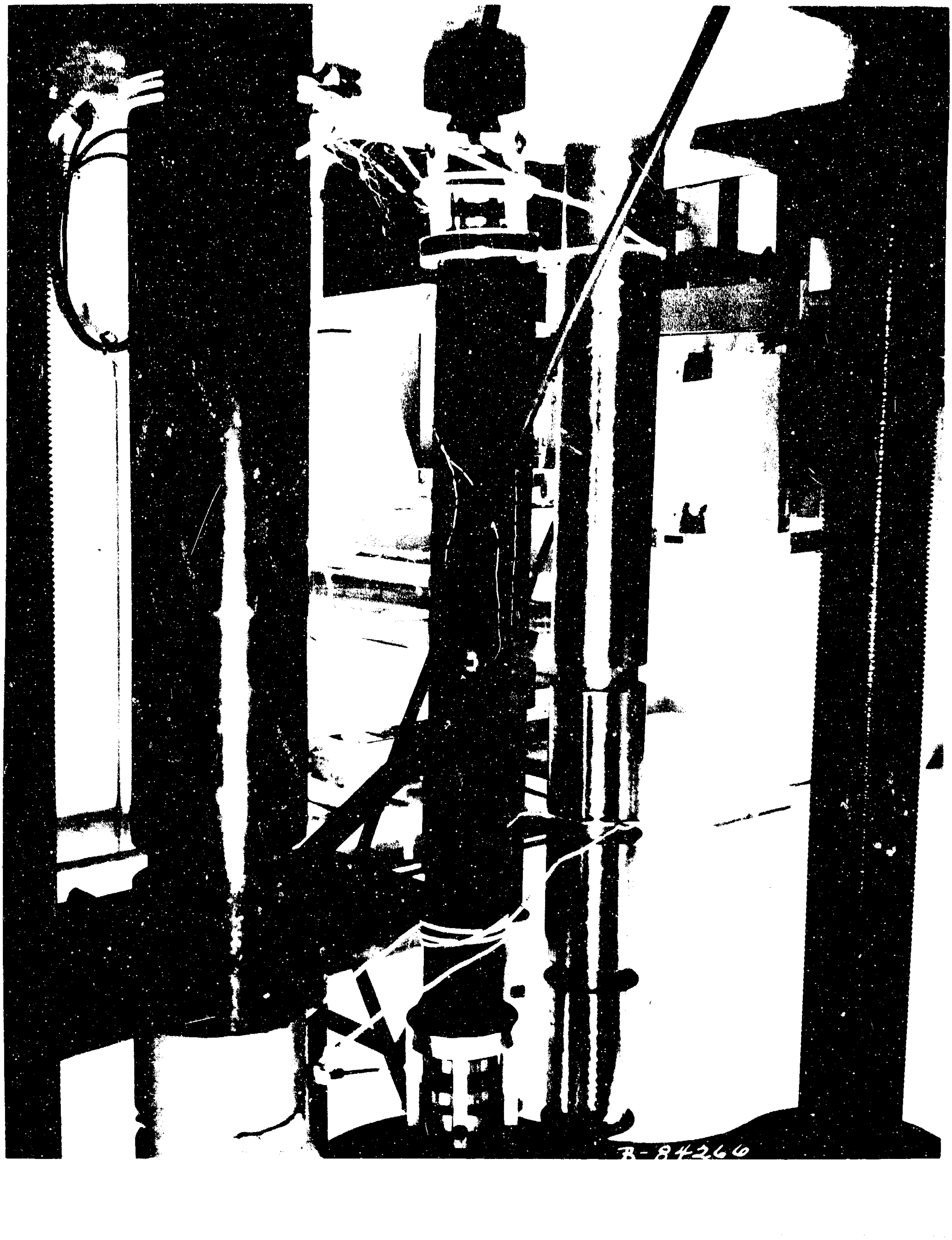

PLOT OF LOAD APPLIED VS LONGITUDINAL STRAIN GAGE
REACTIONS FOR A CONSTANT PRESSURE OF 400 PSI.
LOAD (LBS), P
4250
4000
3750
3500
3250
3000
750
2000
3L
1L
2L
E 150,000
CONSTANT BENDING
-40
-20
0
+20
+40
+60
+80
+100
+120
STRAIN (μ IN./IN. X 10)
FIGURE NO. 3

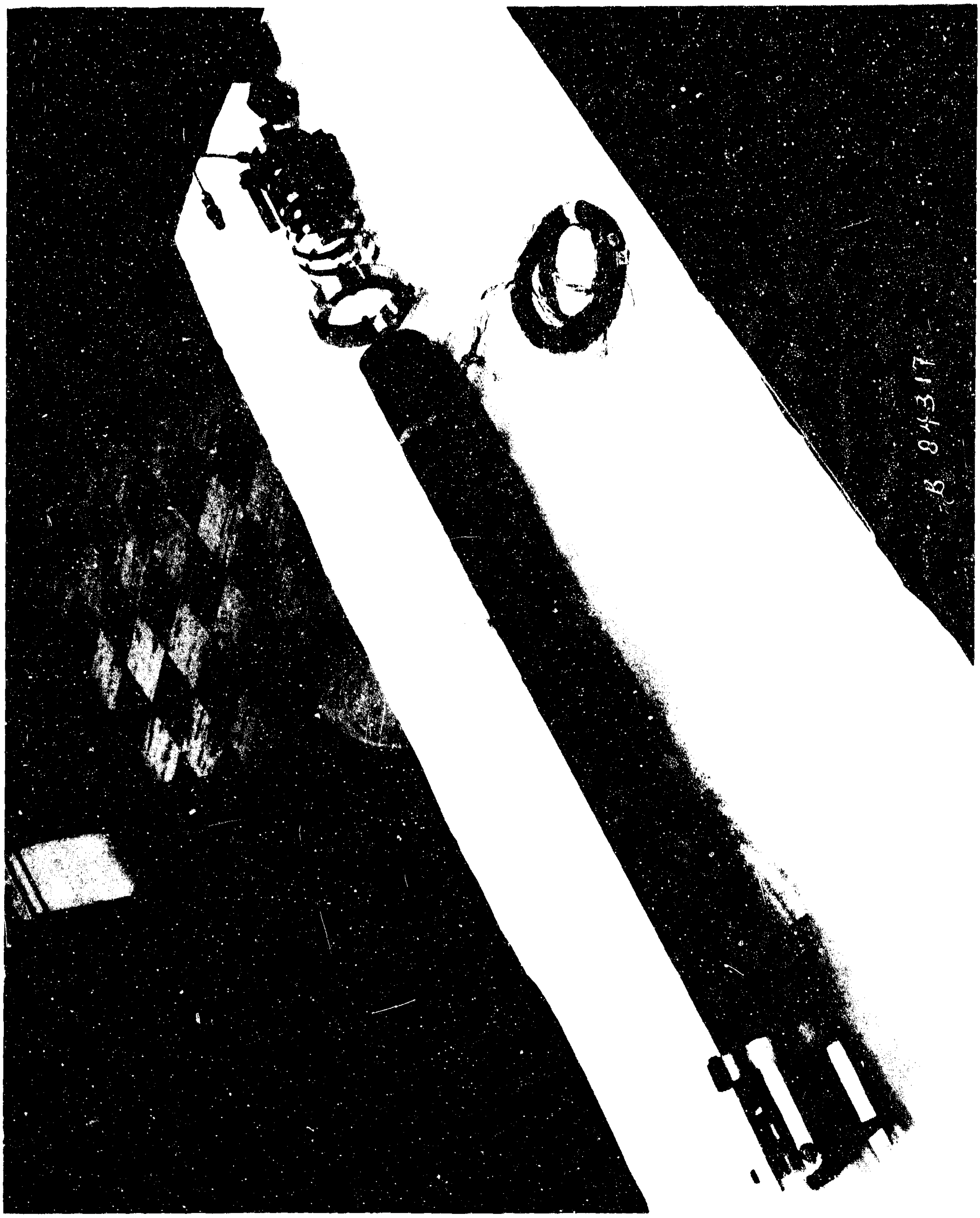

PLOT OF LOAD APPLIED VS LONGITUDINAL
REACTIONS FOR CONSTANT INTERNAL
PRESSURE OF 200 PSI.
E=157,000
CONSTANT BENDING
3L
1L
2L
APPLIED LOAD (LBS), P
2200
2000
1800
1600
1400
1200
1000
800
600
400
200
0
300
1500
2500
-40
-20
0
+20
+40
+60
+80
+100
+120
STRAIN (μIN./IN. X10)
FIGURE NO. 4.


PLOT OF INTERNAL PRESSURE VS TRANSVERSE
STRAIN GAGE REACTIONS
1T
2T
3T
E = pr/t (FOR 3T)
E = (400-100)/200X10^-6 (2) = 3X10^6 lb/in
INTERNAL PRESSURE (psi)
500
400
300
200
100
0
+100
+200
+300
STRAIN (μIN./IN. X10)
FIGURE NO. 5.


1000
900
800
700
600
500
400
300
200
LOAD (LBS)
17
37
37
STRAIN
FIGURE NO. 6


B-84265


B-84266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